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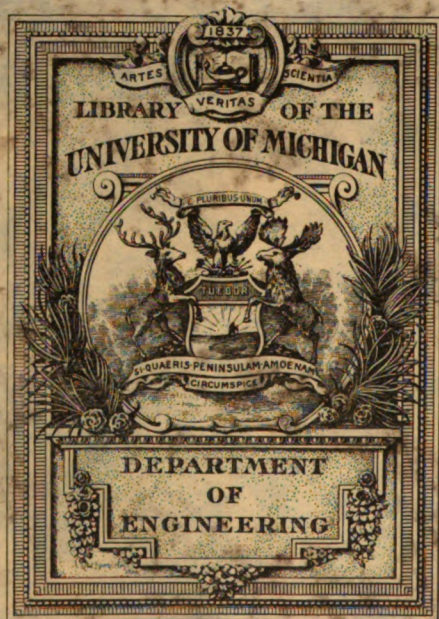
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# THERMODYNAMICS OF THE STEAM TURBINE

*revised by*  
C. H. PEABODY

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MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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## PREFACE

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THIS is an attempt at a systematic treatment of the thermodynamics of the steam-turbine for students in technical schools. Constructive details are considered so far as they are necessary for a proper understanding of the thermodynamic computations or as they are related to such computations.

The methods are in general those that are accepted by steam-turbine designers, but certain methods have been devised by the writer either to make the methods more complete or to provide more rapid and precise determinations of conditions and proportions.

Students undertaking the discussion of steam-turbines must have a sound preparation in general thermodynamics, such as is found in any good textbook on the subject; consequently the introduction gives only an abstract of the properties and computations for steam, mainly for convenience and to avoid ambiguity.

All computations are made by aid of the writer's *Steam and Entropy Tables*; other tables or charts may be substituted if preferred.

It is to be regretted that factors for frictional and other losses are not better known, but in the end those factors must be related to the steam consumption which is pretty well known, and uncertainty as to the distribution of factors for losses can have only a secondary influence.

C. H. P.



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# THERMODYNAMICS OF THE STEAM-TURBINE

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## CHAPTER I

### INTRODUCTION

THOUGH the student of the thermodynamics of the steam-turbine will have as a preparation a thorough course in general thermodynamics, it will be convenient, and will avoid possibility of confusion, to give here an abstract of the properties of steam and of the methods of calculation which depend on those properties. An extensive discussion of the properties of steam and of the construction of tables for computation will be found in the author's *Steam and Entropy Tables*. All the computations in this book are made by aid of those tables, which will be found convenient for the purpose, more especially the temperature-entropy table, which was constructed to facilitate computations for steam-turbines. They will be referred to as the *Tables*.

For engineering purposes steam is generated in a boiler which is partially filled with water, and arranged to receive heat from the fire in the furnace. The ebullition is usually energetic, and more or less water is mingled with the steam; but if there is a fair allowance of steam space over the water there will usually be only a small percentage of water, varying from  $\frac{1}{2}$  to  $1\frac{1}{2}$  per cent. Steam which contains a considerable percentage of water is passed through a separator, which removes almost all of it. Such steam is considered to be approximately dry.

If steam is quite free from water it is said to be dry and saturated; though it is easy to have steam nearly dry it is very difficult to avoid a small amount of moisture.

Steam which is withdrawn from the boiler may be heated to

a higher temperature than that found in the boiler, and is then said to be superheated.

**Saturated Steam.** — All the properties of saturated steam depend on the temperature and may be determined either directly from experiments or by simple computations from those experiments. Table I of the *Tables* gives those properties in English units for each degree Fahrenheit; Table II gives the same properties for each pound pressure. Table III gives properties in both French and English units for each degree Centigrade; it may serve as a conversion table as well as for making computations.

**Thermometric Scales.** — Temperatures are commonly measured with mercurial thermometers which are graduated on the Fahrenheit or Centigrade scales. The Centigrade scale has the zero at freezing-point and the boiling-point is called  $100^{\circ}$  C. The Fahrenheit thermometer has its freezing-point numbered  $32^{\circ}$  F., and its boiling-point  $212^{\circ}$  F. It is clear that

$$t_F = \frac{9}{5} t_C + 32 \quad \text{and} \quad t_C = \frac{5}{9} (t_F - 32).$$

Physicists base their heat measurements on a thermodynamic scale, which is determined from certain theoretical considerations of the properties of gases. The results of experiments on the properties of steam are based on such a scale, and consequently the tables of properties of steam are so also; but for engineering purposes the difference between this scale and the scale of the mercurial thermometer is not important.

**Standard Temperature.** — It is customary to take  $62^{\circ}$  F. as the standard temperature for steam computations; this differs inappreciably from  $15^{\circ}$  C., which is a common standard for physical work.

**Heat Unit.** — The unit for the measurement of heat is the amount of heat required to raise one unit of weight of water one degree from the standard temperature.

The British thermal unit is the amount of heat required to raise the temperature of one pound of water from  $62^{\circ}$  to  $63^{\circ}$  F.

The calorie is the amount of heat required to raise the temperature of one kilogram of water from  $15^{\circ}$  to  $16^{\circ}$  C.

**Mechanical Equivalent of Heat.** — If mechanical energy or work is transformed into heat and applied to heating water, it will be found that 778 foot-pounds of work will be required to heat one pound of water from 62° to 63° F.; in other words, one B.T.U. is equivalent to that number of foot-pounds. This is known as the mechanical equivalent of heat. In the French system of units the mechanical equivalent of one calorie is 427 meter-kilograms.

**Specific Heat** is the number of thermal units required to raise a unit of weight of a given substance one degree of temperature. The specific heat of water at standard temperature is unity, and any specific heat is essentially a ratio.

**Specific Heat of Water.** — The specific heat of water is slightly greater than unity at freezing-point; it diminished to unity at 62° F., it becomes a minimum at about 100° F., and increases again at higher temperatures. The mean specific heat between freezing and boiling points is nearly unity. For rough approximations it can be assumed to be unity for all temperatures less than 212° F.

**Heat of the Liquid.** — The heat required to raise one unit of weight of any liquid from freezing-point to a given temperature is called the heat of the liquid at that temperature. It is represented by  $q$ .

The heats of the liquid in Tables I, II, and III were obtained by a combination of computation and graphical integration allowing for the varying specific heat of water.

**Heat of Vaporization.** — If a unit of weight of a liquid be at a certain temperature and subject to the corresponding pressure, then the amount of heat required to vaporize it into dry saturated vapor at that temperature and against that pressure is called the *heat of evaporation*. Below boiling-point the heats of vaporization are computed by the equations:

$$\text{English units} \quad r = 141.124 (689 - t)^{0.31249} \quad . \quad . \quad . \quad (1)$$

$$\text{French units} \quad r = 94.210 (365 - t)^{0.31249} \quad . \quad . \quad . \quad (2)$$

**Total Heat.** — The amount of heat required to raise a unit of weight of liquid from freezing-point to a given temperature and to vaporize it into dry saturated vapor against the corresponding temperature is called *total heat*.

This quantity is clearly equal to the sum of the heat of the liquid and the heat of vaporization; it can be represented by

$$H = r + q.$$

Above boiling-point the total heat is given by the equations:

English units

$$H = 1150 + 0.3745(t - 212) - 0.00055(t - 212)^2 \quad . \quad . \quad (3)$$

French units

$$H = 638.9 + 0.3745(t - 100) - 0.00099(t - 100)^2 \quad . \quad . \quad (4)$$

**Specific Pressure.** — It is customary to develop theoretical thermodynamic equations with the specific pressure expressed in pounds per square foot for English units. Engineers habitually express pressures in pounds per square inch.

For French units, specific pressures are expressed in kilograms per square meter. Engineers use kilograms per square centimeter, and on the other hand physicists commonly express pressures in millimeters of mercury.

**Pressure of Saturated Steam.** — Recent determinations of the pressure of saturated steam have been made by Holborn and Henning with all the resources of modern physical methods. Their results are set down in Table III; as their results extend to 205° C., only the pressures above that temperature were extrapolated graphically.

The pressures in Table I were derived from those in Table III, allowing for changes of units. The temperatures of Table II were obtained by interpolation in Table I.

**Normal Pressure.** — The normal pressure of the atmosphere may be taken as

- 14.7 pounds per square inch;
- 2116 pounds per square foot;
- 760 millimeters of mercury;
- 1.0333 kilograms per square centimeter;
- 1033.3 kilograms per square meter.

**Laws of Thermodynamics.** — Theoretical thermodynamics is based on two propositions or laws known as the *first* and *second laws*. The first law is stated on page 3, under the heading Mechanical Equivalent of Heat.

The second law cannot be stated so briefly and satisfactorily; it may perhaps be best represented by one of its consequences, which can be written

$$e = \frac{T - T'}{T}, \quad . . . . . (5)$$

where  $e$  is the efficiency of an ideal perfect heat engine and  $T$  and  $T'$  are absolute temperatures at which the engine receives and rejects heat.

For our present purpose it may be sufficient to define the absolute temperature by the expressions

$$T = t + 273^{\circ} \text{ C.}$$

$$T = t + 459^{\circ}.5 \text{ F.}$$

**Specific Volume of Saturated Vapor.** — From the extreme difficulty of direct experimental determination of the specific volume of saturated vapor it has been customary to compute this property by aid of the equation

$$s = u + \sigma = \frac{r}{AT} \frac{1}{\Delta p} + \sigma \quad . . . . . (6)$$

$\Delta t$

which is derived by the application of the two laws of thermodynamics.

In this equation  $r$ ,  $T$ , and  $A$  are respectively the heat of vaporization, the absolute temperature, and the reciprocal of the mechanical equivalent of heat.  $\Delta p$  is a small increase of pressure due to a small increase of temperature  $\Delta t$ . Finally  $\sigma$  is the specific volume of water 0.016.

Now  $s$  is the volume in cubic feet occupied by one pound of steam, and  $\sigma$  is the specific volume of the liquid. The increase of volume due to vaporization is

$$u = s - \sigma.$$

In the French system of units the specific volume of water is 0.001 of a cubic meter per kilogram.

**Internal and External Latent Heat.** — The heat of vaporization overcomes external pressure and changes the state from liquid to vapor at constant temperature and pressure. The specific volume of saturated steam being  $s$ , and the specific volume of water being  $\sigma$ , then the increase of volume due to vaporization is  $u$ , and the external work is

$$p(s - \sigma) = pu. \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The heat equivalent of the external work is

$$Apu,$$

where  $A$  is the reciprocal of the mechanical equivalent of heat.

That part of the heat of vaporization which is not used in doing external work is considered to be used in changing the state from liquid to vapor. This work required to change the molecular arrangement is called disgregation work, and is represented by

$$p = r - Apu. \quad . \quad . \quad . \quad . \quad . \quad (8)$$

**Quality or Dryness Factor.** — All the properties of saturated steam, such as pressure, volume, and heat of vaporization, depend on the temperature only, and can be taken from Tables I, II, or III.

Many of the problems met in engineering deal with mixtures of liquid and vapor, such as water and steam. In such problems it is convenient to represent the proportions of water and steam by a variable known as the quality or dryness factor; this factor,  $x$ , is defined as that portion of each pound of the mixture which is steam; the remnant,  $1 - x$ , is consequently water.

**Specific Volume of Wet Steam.** — If a pound of a homogenous mixture of water and steam is  $x$  part steam, then the specific volume may be represented by

$$v = xs + (1 - x)\sigma = xu + \sigma.$$

**Intrinsic Energy.** — When heat is applied to a substance, a part is expended in increasing the temperature, a part is required to do the external work, and the remainder is considered to be

used up in changing the molecular arrangement or condition. It has been seen that these three portions can be separated for saturated vapor; they are represented by  $q$ ,  $Apu$ , and  $\rho$ . In some cases the first and last cannot be separated and must be treated together; in any case it is convenient to consider them together. The mechanical equivalent of their sum is called the intrinsic energy and may be represented by

$$E = \frac{1}{A}(\rho + q). \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

If only a portion of the liquid is vaporized the external work and the disgregation work may be obtained by multiplying the proper quantities by the dryness factor, and the heat equivalents will be

$$Axpu \text{ and } x\rho.$$

In such case the intrinsic energy is

$$E = \frac{1}{A}(x\rho + q). \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

**Entropy.** — In the discussion of steam engines it is convenient to begin by considering the way in which steam would behave if the cylinder were made of non-conducting material. Afterwards the effect of the actual material can be investigated. The expansion line which an indicator would draw under such conditions is called an adiabatic line. Calculations for adiabatic changes of steam can be made by aid of a special function devised for the purpose and called *entropy*. It is sufficient for our present purpose to consider that entropy can be expressed numerically, and that the numerical values enter into the calculation of certain engineering problems.

It is customary to represent entropy in general by  $\phi$ , but entropy may be represented by  $\theta$  in dealing with a liquid.

To calculate the increase of entropy during any operation we may divide the heat added by the absolute temperature at which it is added. If the heat is added at a varying temperature, an approximation may be had by breaking the heat into

small portions and dividing each by the mean temperature and then summing up.

Expressed in the forms of the infinitesimal calculus

$$\phi - \phi_0 = \int \frac{dQ}{T}, \quad \dots \dots \dots (11)$$

in which  $dQ$  is the heat added and  $T$  is the absolute temperature at which it is added.

Since the adiabatic expansion line for a non-conducting cylinder is drawn at constant entropy it is frequently called an *isentropic* line. There are some adiabatic operations which are not isentropic, — for a discussion of this subject reference should be made to a textbook on thermodynamics, — in this work the terms will be used interchangeably and restricted to the case mentioned, that is, to the equivalent of expansion in a non-conducting cylinder.

**Entropy of Vaporization.** — If a pound of water at the temperature  $t$  (or absolute temperature  $T$ ) is partially vaporized the heat expended is  $xr$ . The increase of entropy due to vaporization is

$$\phi - \phi_0 = \frac{xr}{T} = x \frac{r}{T} \quad \dots \dots \dots (12)$$

In Tables I, II, and III the values of  $\frac{r}{T}$  are given for each degree or each pound.

**Entropy of the Liquid.** — When water is heated the specific heat varies and the heat is added at a varying temperature. While an approximation can be had by breaking up the heat into small parts as indicated in a preceding paragraph, a satisfactory determination of the entropy of the liquid can be made only by aid of the integral calculus, which gives

$$\theta = \int \frac{dq}{T} = \int \frac{cdt}{T} \quad \dots \dots \dots (13)$$

It is sufficient to say that taking account of the variation of the specific heat of water, the values of  $\theta$  in Tables I, II, and III were determined by a combination of numerical computation and graphical integration.

**Entropy of a Mixture of a Liquid and its Vapor.** — The increase of entropy due to heating a unit of weight of a liquid from freezing-point to the temperature  $t$  and then vaporizing  $x$  portion of it is

$$\theta + \frac{x\tau}{T}$$

where  $\theta$  is the entropy of the liquid,  $\tau$  is the heat of vaporization, and  $T$  is the absolute temperature;  $\theta$  and  $\frac{\tau}{T}$  may be taken from the *Tables*.

For another condition determined by  $x_1$  and  $t_1$  we shall have for the increase of entropy above freezing-point,

$$\frac{x_1\tau_1}{T_1} + \theta_1.$$

The change of entropy in passing from one state to another is

$$\phi_1 - \phi_1 = \frac{x\tau}{T} + \theta - \frac{x_1\tau_1}{T_1} - \theta_1. \quad \dots \quad (14)$$

**Adiabatic Equation for a Liquid and its Vapor.** — During an adiabatic change the entropy is constant, so that the preceding equation gives

$$\frac{x_1\tau_1}{T} + \theta_1 = \frac{x_2\tau_2}{T_2} + \theta_2. \quad \dots \quad (15)$$

When the initial state, determined by  $x_1$  and  $t_1$  or  $p_1$ , is known and the final temperature,  $t_2$ , or the final pressure,  $p_2$ , the final value,  $x_2$ , can be calculated by this equation. The initial and final volumes can be found by the equations

$$v_1 = x_1u_1 + \sigma \quad \text{and} \quad v_2 = x_2u_2 + \sigma.$$

Tables of the properties of steam give the specific volume  $s$ , but

$$s = u + \sigma.$$

The value of  $\sigma$  for water is 0.016 using English units, or 0.001 for French unit.

*For example*, one pound of dry steam at 100 pounds absolute has the following properties found in Table II, *Steam and Entropy Tables*,

$$\frac{\tau_1}{T_1} = 1.1273; \theta = 0.4748; s_1 = 4.432; x_1 = 1.$$

If the final pressure is 15 pounds absolute,

$$\frac{r_2}{T_2} = 1.4409; \theta_2 = 0.3140; s_2 = 26.28,$$

whence

$$1.1273 + 0.4748 = 1.4409 x_2 + 0.3140 \\ \therefore x_2 = 0.8939.$$

The initial and final volumes are

$$v_1 = s_1 = 4.432$$

$$v_2 = x_2 v_1 + \sigma = 0.8939 (26.28 - 0.016) + 0.016 = 23.49.$$

Such a problem cannot be solved inversely, that is, we cannot assume a final volume and determine directly the corresponding pressure. The temperature-entropy table to be explained later will give an approximate solution directly, and an exact solution by interpolation.

**External Work During Adiabatic Expansion.** — Since no heat is transmitted during an adiabatic expansion, all the intrinsic energy lost is changed into external work, so that

$$W = E_1 - E_2 = \frac{1}{A} (x_1 \rho_1 + q_1 - x_2 \rho_2 - q_2). \quad (16)$$

*For example*, the external work of one pound of dry steam expanding adiabatically from 100 pounds to 15 pounds absolute is

$$W = 778 (805.7 + 298.5 - 0.8939 \times 896.2 - 181.3)$$

$$W = 121.8 \times 778 = 94,760 \text{ foot-pounds.}$$

Attention should be called to the unavoidable defect of this method in that it depends on taking the difference of quantities which are of the same order of magnitude. The above calculation appears to give four places of significant figures while, as a matter of fact, the total heat,  $H$ , from which  $\rho$  is derived, is affected by a probable error of  $\frac{1}{1000}$  or perhaps more. Both quantities,

$$x_1 \rho_1 + q_1 \quad \text{and} \quad x_2 \rho_2 + q_2,$$

have a numerical value somewhere near 1000, and an error of  $\frac{1}{1000}$  is nearly equivalent to one thermal unit. The probable error

of the above calculation is about three-fourths of one per cent. Had the final pressure been one pound absolute, the probable error would have been one-third of one per cent. But for a narrow range of pressure the error would be larger. This should be borne in mind in the use of diagrams, like the temperature-entropy diagram or Mollier's diagram.

**Heat Contents.** — The heat required to raise one pound of water from freezing-point to a given temperature  $t$  corresponding to a pressure  $p$ , and to vaporize a part  $x$  at that pressure, is represented by

$$xr + q;$$

this quantity may be called the heat contents.

**Superheated Steam.** — A dry and saturated vapor, not in contact with the liquid from which it is formed, may be heated to a temperature greater than that corresponding to the given pressure for the same vapor when saturated; such a vapor is said to be *superheated*.

The equation for superheated steam, using French units, is

$$pv = BT - p(1 + ap) \left[ C \left( \frac{373}{T} \right)^3 - D \right]. \quad (17)$$

The pressure  $p$  is in kilograms per square meter,  $v$  is specific volume in cubic meters, and  $T$  is the absolute temperature by the Centigrade thermometer. The constants are:

$$B = 47.10, \quad a = 0.000002, \quad C = 0.031, \quad D = 0.0052.$$

In the English system of units the pressures being in pounds per square inch, the volumes in cubic feet, and the temperature in the Fahrenheit scale,

$$pv = 0.5962 T - p(1 + 0.0014p) \left( \frac{150,300,000}{T^3} - 0.0833 \right). \quad (18)$$

The specific heats in the Entropy Table were determined by aid of this equation. The labor of calculation is principally in the reduction of the term containing  $T^3$ ; Table XV of the *Tables* facilitates computations by equation (18).

**Specific Heat of Superheated Steam.** — The specific heat of superheated steam varies in a marked manner, depending on the pressure and the degree of superheat.

The following table gives mean specific heats from saturation temperature to the temperatures given at the left of the table:

SPECIFIC HEAT OF SUPERHEATED STEAM

$p$ Kg. per sq. cm.		1	2	4	6	8	10	12	14	16	18	20
$p$ Lbs. per sq. in.		14.2	28.4	56.9	85.3	113.8	142.2	170.6	199.1	227.5	156.0	284.4
$t_s$ Cent.		99°	120°	143°	158°	169°	179°	187°	194°	200°	206°	211°
$t_s$ Fahr.		210°	248°	289°	316°	336°	350°	368°	381°	392°	403°	412°
Fahr.	Cent.											
212°	100°	0.463	...	...	...	...	...	...	...	...	...	...
302°	150°	0.462	0.478	0.515	...	...	...	...	...	...	...	...
392°	200°	0.462	0.475	0.502	0.530	0.560	0.597	0.635	0.677	...	...	...
482°	250°	0.463	0.474	0.495	0.514	0.532	0.552	0.570	0.588	0.609	0.635	0.664
572°	300°	0.464	0.475	0.492	0.505	0.517	0.530	0.541	0.550	0.561	0.572	0.585
662°	350°	0.468	0.477	0.492	0.503	0.512	0.522	0.529	0.536	0.543	0.550	0.557
752°	400°	0.473	0.481	0.494	0.504	0.512	0.520	0.526	0.531	0.537	0.542	0.547

The construction of this table is readily understood from the following example: *Required* the heat needed to superheat a kilogram of steam at 4 kilograms per square centimeter from saturation to 300° C. The saturation temperature (to the nearest degree) is 143° C.; so that the steam at 300° is superheated 157°, and for this the heat required is

$$157 \times 0.492 = 77.2 \text{ calories.}$$

**Total Heat of Superheated Steam.** — In the solution of problems that arise in engineering it is convenient to use the total amount of heat required to raise one pound of water from freezing-point to the temperature of saturated steam at a given pressure and to vaporize it and to superheat it at that pressure to a given temperature. This total heat may be represented by the expression

$$H = q + r + c_p(t - t_s), \quad \dots \quad (19)$$

when  $t$  is the temperature of the superheated steam,  $t_s$  is the temperature of saturated steam at a given pressure  $p$ , and  $q$  and  $r$  are the corresponding heat of this liquid and the heat of vaporization; finally,  $c_p$  is the mean specific heat from the preceding table.

The total heats for superheated steam in the temperature-entropy table were obtained from a table (not given here) which gives specific heats at certain temperatures and pressures. The results differ but slightly from values that can be found by the preceding table.

**Entropy of Superheat.** — By the entropy of superheated steam is meant the increase of entropy due to heating water from freezing-point to the temperature of saturated steam at a given pressure, and to the vaporization, and to the superheating at that pressure. This operation may be represented as follows:

$$\theta + \frac{r}{T_s} + \int_{T_s}^T \frac{c_p dt}{T}, \quad . . . . . (20)$$

in which  $T$  is the absolute temperature of the superheated steam at the given pressure;  $\theta$  and  $\frac{r}{T_s}$  can be taken from Table I of the *Tables*. The last term was obtained for the temperature-entropy table by graphical integration.

**Temperature-Entropy Table.** — This table was made to facilitate the solution of problems involving adiabatic action of steam and some other problems. The properties given in the table are:

Moist steam:

Quality,  $x$ ; the portion of a pound which is steam.

Heat contents,  $x_1 r_1 + q$ .

Specific volume,  $v = xu + \sigma$ .

Superheated steam:

Quality,  $t - t_s$ ; the number of degrees of superheating.

Heat contents,  $r + q + c_p (t - t_{sat})$ .

Specific volume,  $v$ .

## CHAPTER II

### STEAM-NOZZLES

THE flow of steam through an orifice or a nozzle is an irreversible process during which some of the available heat is changed into kinetic energy of motion, for which allowance must be made. The following is the customary method of establishing the fundamental equation: Suppose that a fluid is flowing from the larger pipe *A* into the pipe *B*; there will be an increase of velocity with a reduction in pressure. In order to

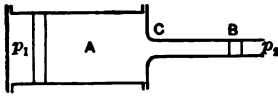


FIG. 1.

make the conception more complete let it be supposed that there is a frictionless piston in each cylinder; the piston in *A* exerts the pressure  $p_1$  on the fluid in front of it; and the piston in *B* has on it the fluid pressure  $p_2$ . Each unit of weight of fluid passing from *A* through the orifice has the work  $p_1 v_1$  done on it, while each pound entering the cylinder *B* does the work  $p_2 v_2$ . The assumption of pistons is merely a matter of convenience, and if they are suppressed the same condition with regard to external work will hold.

Each pound of steam in the cylinder *A* moving toward the orifice carries the intrinsic energy

$$E_1 = \frac{1}{A} (x_1 \rho_1 + q_1).$$

If the velocity of the steam in that cylinder is  $V_1$  feet per second the kinetic energy per pound of steam is

$$\frac{V_1^2}{2g};$$

when  $g$  is the acceleration due to gravity (32.16 feet).

In like manner the steam in the cylinder  $B$  will carry the intrinsic energy

$$E_2 = \frac{1}{A} (x_2 \rho_2 + q_2)$$

and will have the kinetic energy

$$\frac{V_2^2}{2g}$$

per pound of steam.

Summing up the external work, the intrinsic energy and the kinetic energy for each cylinder, and equating the results, we have

$$p_1 v_1 + E_1 + \frac{V_1^2}{2g} = p_2 v_2 + E_2 + \frac{V_2^2}{2g}; \quad . . . \quad (1)$$

this is the general fundamental equation depending only on the first law of thermodynamics, which asserts that heat may be converted into work at the fixed ratio of 778 foot-pounds per thermal unit.

No account is taken in this equation of radiation or other external losses, nor of friction of the steam along the pipe. The effect of radiation or other heat losses will always be inconsiderable for orifices and nozzles; the method of allowing for friction will be discussed later.

If the velocity of the steam in the cylinder  $A$  is small its influence may be neglected and  $V_1$  may be omitted from equation (1); at the same time the subscript may be dropped from the letter representing the velocity in  $B$ . Then

$$\frac{V^2}{2g} = E_1 - E_2 + p_1 v_1 - p_2 v_2. \quad . . . \quad (2)$$

**Flow of Steam.** — For steam we may replace  $E_1$  and  $E_2$  by values given above, so that the equation becomes

$$\frac{V^2}{2g} = \frac{1}{A} (x_1 \rho_1 + q_1 - x_2 \rho_2 - q_2) + p_1 v_1 - p_2 v_2. \quad . . \quad (3)$$

Now

$$v_1 = x_1 u_1 + \sigma \quad \text{and} \quad v_2 = x_2 u_2 + \sigma.$$

†

Substituting in equation (3) and transposing

$$A \frac{V^2}{2g} = q_1 - q_2 + x_1 \rho_1 + A p_1 x_1 u_1 - A p_2 x_2 u_2 + A \sigma (p_1 - p_2).$$

But

$$\rho + A p u = r.$$

$$\therefore A \frac{V^2}{2g} = x_1 r_1 + q_1 - x_2 r_2 - q_2, \quad . \quad . \quad . \quad . \quad (4)$$

provided that the term containing  $\sigma$  be omitted because it is small.

It is customary to compute the unknown value of  $x_2$  by the equation for entropy,

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2, \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which was deduced by aid of the second law of thermodynamics. There is some theoretical criticism of the use of equation (5) in this place; but experiments on orifices and nozzles show that the weight of steam actually discharged differs from the amount computed by a small percentage which can be conveniently allowed for by means of a friction factor. Again, the right-hand member of equation (4) represents the heat changed into work for the Rankine cycle for a steam-engine, as will appear on page 35, which indicates that the flow of steam in a nozzle differs from its action in a cylinder, mainly in that the work done is applied to increasing the kinetic energy of the steam instead of driving the piston.

The weight of steam that will pass through an orifice having an area of a square foot may be found by the formula

$$w = \frac{aV}{v_2} = \frac{aV}{x_2 u_2 + \sigma} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The equations (4), (5), and (6) may be solved by aid of Table I or II of the *Tables*; or more conveniently, the quantities  $x_1 r_1 + q_1$ ,  $x_2 r_2 + q_2$  and  $x_2 u_2 + \sigma$  may be taken from the temperature-entropy table. In the ordinary use of that table it

will be sufficient to take the nearest temperature; should, however, the conditions come half way between two temperatures in that table, it may be advantageous to interpolate. For calculating velocity the nearest entropy column may always be chosen; for while the values of

$$x_1 r_1 + q_1$$

may change appreciably from column to column, the value of

$$x_1 r_1 + q_1 - (x_2 r_2 + q_2)$$

for a range of temperature  $t_1 - t_2$  will change very little.

*Example.*—Required the flow through an orifice one inch in diameter under a difference of pressure from 100 pounds to 75 pounds by the gauge. Assuming the pressure of the atmosphere to be 14.7 pounds the absolute pressures are 114.7 and 89.7 pounds. The problem can be solved by reference to pages 81 and 83 of the temperature-entropy table, entropy 1.58, from which the following properties are taken:

$$\begin{aligned} p_1 &= 114.8, \quad t_1 = 338^\circ, \quad x_1 r_1 + q_1 = 1179.8 \\ p_2 &= 89.6, \quad t_2 = 320^\circ, \quad x_2 r_2 + q_2 = \frac{1160.0}{19.8} \quad v_2 = 4.775 \end{aligned}$$

$$V = \sqrt{778 \times 2 \times 32.16 \sqrt{66.6}} = 223.7 \times 4.45 = 995.$$

The area of an orifice one inch in diameter is 0.7854 of a square inch or 0.005454 of a square foot. Consequently this discharge will be

$$w = \frac{0.005454 \times 995}{4.775} = 1.135$$

pounds per second or 4080 pounds per hour.

**Flow of Superheated Steam.**—There is no convenient expression for the intrinsic energy of superheated steam, consequently the velocity cannot be deduced directly from equation (2), page 15; but an equation for practical application can be obtained by adding to equation (4) a term to allow for superheating, as is done in equation (19), page 12, for computing

the total heat of superheated steam. This gives, for a case when the steam becomes wet in the orifice,

$$\frac{V^2}{2g} = c_p (t_1 - t_s) + r_1 + q_1 - x_2 r_2 - q_2 \dots \quad (7)$$

But the temperature-entropy table gives a value for the positive part of the right-hand member (determined by a more precise method) under the title of *heat contents* for superheated steam. Consequently the computation for flow through an orifice can be made with the same facility for superheated steam as for saturated steam; and the solution can be made with the same facility should the steam remain superheated in the orifice. In such case equation (7) may be considered to have the form

$$\frac{V^2}{2g} = (\text{heat contents})_1 - (\text{heat contents})_2 = C_1 - C_2 \quad (7a)$$

where  $C$  stands for the heat contents of the temperature-entropy table.

*Example.* — Suppose that in the preceding example the steam is superheated about  $42^\circ$ ; then the solution can be made in the column for entropy 1.62, page 94, of the *Tables* which gives

$$\begin{array}{ll} p_1 = 114.8, \text{ heat contents} = 1212.4 & \\ p_2 = 89.6 & \frac{1191.2}{21.2} \quad v_2 = 5.018 \end{array}$$

$$V = 223.7 \times \sqrt{21.2} = 223.7 \times 4.60 = 1029.$$

The discharge in pounds per second becomes

$$w = \frac{0.005454 \times 1029}{5.018} = 1.12$$

pounds per second, or 4030 pounds per hour.

**Rankine's Equations.** — The following equations may be used for approximate computations of discharge of saturated steam by orifices. Let  $p_a$  be the absolute pressure of the atmosphere or of the chamber into which steam is discharged, and let  $p_1$  be the absolute pressure at entrance, both in pounds per square

inch, and let  $a$  be the orifice in *square inches*. Then the discharge in pounds per second will be:

$$\begin{aligned} \text{For } p_1 = \text{or } > \frac{5}{3} p_a, & \quad w = a \frac{p_1}{70} \\ p_1 < \frac{5}{3} p_a & \quad w = 0.029 a [p_a (p_1 - p_a)]^{\frac{1}{2}} \end{aligned}$$

The error of these equations is seldom more than two per cent. They may also be applied to calculating the flow from diverging nozzles.

**Grashoff's Formula.** — For pressures exceeding five-thirds the back-pressure the following equation may be used for computing the discharge of saturated steam:

$$w = 0.0165 a p_1^{0.97},$$

in which  $p_1$  is the absolute pressure in pounds per square inch,  $a$  is the area in square inches, and  $w$  is the discharge in pounds per second. This equation gains interest from the fact that Rateau has used it for rating the flow through thin plates and converging nozzles for all ratios of entrance pressure to back-pressure, which latter may be the pressure of the atmosphere or of the chamber into which the nozzle discharges.

RATEAU'S FACTORS

Ratio of back-pressure to pressure at entrance.	Ratio of actual to computed discharge.	
	Converging orifice.	Orifice in thin plate.
0.95	0.45	0.30
0.90	0.62	0.42
0.85	0.73	0.51
0.80	0.82	0.58
0.75	0.89	0.64
0.70	0.94	0.69
0.65	0.97	0.73
0.60	0.99	0.77
0.55	...	0.80
0.45	...	0.82
0.40	...	0.83

**Forms of Orifices and Nozzles.** — For the sake of estimating the flow of steam through a pipe supplying auxiliary machinery

it is sometimes convenient to use an orifice in a thin plate that can readily be slipped in between the flanges of a joint in the pipe. In such case the fall of pressure should be 15 pounds or more, and the pressures must be measured on both sides of the diaphragm. It has been proposed to use orifices in thin plates for turbines which have many pressure stages, but better results can be had with converging nozzles.

Fig. 2 shows the form of converging nozzle used for the Rateau pressure-compound turbines which have many stages, so



FIG. 2.

that the ratio of the entrance pressure to the pressure in the chamber to which the nozzle discharges is less than five-thirds. Rateau's large experience in designing and testing turbines gives

importance to this form of nozzle.

Fig. 3 shows the form of nozzle used for the de Laval turbine. The entrance is rounded with a radius about equal to the diameter of the throat of the nozzle. There is then a short

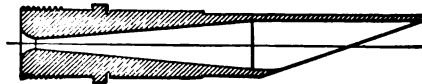


FIG. 3.

cylindrical part, the length being less than the diameter of the throat, which is gently rounded into a straight conical expansion tube. The taper of the tube is about one in twelve. Expanding nozzles are commonly made on this pattern and probably give as good efficiency as any other form.

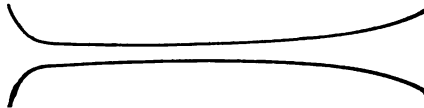


FIG. 4.

Fig. 4 shows a nozzle which is computed to give uniform acceleration along the axis of the nozzle. It will be seen that

it is not very different from the de Laval nozzle illustrated by Fig. 3.

**Friction Head.** — It is customary to compute the velocity of flow of water through a pipe by the equation

$$V = \sqrt{2gh} \text{ or } h = \frac{V^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where  $h$  is the head in feet of the water producing the velocity, and  $g$  is the acceleration due to gravity. The additional resistance due to a bend, a valve, or other obstruction, is allowed for by a term proportional to the square of the velocity and affected by a factor  $K$ , which must be determined experimentally. The head is then given by the equation

$$h = \frac{V^2}{2g} + K \frac{V^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad (9)$$

This added term is sometimes called the head due to the resistance, and the equation is then written

$$h = \frac{V^2}{2g} + h' \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Rearranging, we may write

$$\frac{V^2}{2g} = h - h' = h \left( 1 - \frac{h'}{h} \right) = h(1 - \gamma), \quad . \quad . \quad . \quad (11)$$

where  $\gamma$  represents that part of the entire head that must be allowed for overcoming the resistance.

This method has been adapted to computations for flow through orifices when allowance is made for friction by replacing  $h$  by the mechanical equivalent of the available heat for adiabatic action.

Referring to the discussion of the velocity of flow of steam, saturated or superheated, on pages 15 and 17, we may adopt for the general form for the velocity of steam from a nozzle,

$$A \frac{V^2}{2g} = (\text{heat contents})_1 - (\text{heat contents})_2,$$

or, using the letter  $C$  for heat contents,

$$A \frac{V^2}{2g} = C_1 - C_2. \quad \dots \quad (12)$$

If allowance is made for friction in the nozzle the equation may be written

$$\begin{aligned} V &= \sqrt{\frac{2g}{A}} \sqrt{(C_1 - C_2)(1 - y)} \\ &= 223.7 \sqrt{(C_1 - C_2)(1 - y)}, \quad \dots \quad (13) \end{aligned}$$

using  $A = 778$  and  $g = 32.16$ .

This equation may be used for moist steam or superheated steam, taking the heat contents from the temperature-entropy table. With somewhat less precision the heat contents can be taken from Mollier's diagram found at the back of this book.

**Mollier's Diagram.**—For convenience in the determination of flow of steam through orifices and nozzles (and for the solution of other problems) Mollier devised a diagram in which heat contents are plotted as ordinates on entropies for abscissæ. A great advantage is that the difference of heat contents can be measured off directly on the edge of a strip of paper and transferred to a special scale on which the velocity can be read in feet per second. This statement will be evident on reference to the Mollier diagram and the example of its use given with the diagram. The degree of precision will be sufficient when the drop of pressure is large, but not for moderate or small drops of pressure.

**Values for  $y$ .**—Though there have been many experiments on the flow of steam through orifices and nozzles, the value for  $y$  cannot be quoted with the degree of precision that could be desired. Experiments on small orifices 0.1 to 0.2 of an inch in diameter with expansion to a vacuum give  $y$  varying from 0.10 to 0.15. Though the nozzles for steam-turbines are commonly larger than experimental nozzles it is probable that it is wise to take  $y$  to vary from 0.05 to 0.10, to

allow for imperfections in manufacture or roughening during service.

Though in itself the information concerning the friction factor is unsatisfactory, the error in computation of discharge of steam, or conversely of the proper dimensions of nozzles, cannot be serious, because the flow is affected by the function

$$\sqrt{1 - y},$$

and if  $y$  be taken in succession 0, 0.05, 0.10, and 0.15, the value of this function varies as

$$\begin{aligned}\sqrt{1 - 0} &= 1; & \sqrt{1 - 0.05} &= 0.97; & \sqrt{1 - 0.10} &= 0.95; \\ & & \sqrt{1 - 0.15} &= 0.92;\end{aligned}$$

so that the entire variation is only 8 per cent and a conservative choice should not be subject to a greater error than 2 per cent.

**Pressure in the Throat.** — In order to proceed with the design of an expanding nozzle it is necessary to know the pressure in the throat. This pressure is shown by experiments to be 0.58 of the absolute pressure at admission when the back-pressure is not more than three-fifths of the internal pressure.

**Incomplete or Excessive Expansion.** — It has been shown by experiments that nozzles which are correctly designed for complete expansion from a given pressure  $p_1$  at entrance to a pressure  $p_2$  at exit will give expected results very nearly, including the computed discharge. In particular, if the pressures along the tube are measured by small side orifices (or otherwise) the fall of pressures is that which should be expected. If a nozzle designed for a pressure  $p_2$  is used with the proper entrance pressure  $p_1$ , but is allowed to discharge into a chamber having a pressure  $p_3$  less than  $p_2$ , the fall of pressures in the nozzle will be normal, and the pressure will continue to fall in the jet beyond the tube. In such case the expansion is said to be incomplete. The jet beyond the tube tends to break up at its boundary and its action will be less efficient in a turbine, but a nozzle which gives nearly complete expansion will be almost as good as though the expansion were complete.

If, however, such a nozzle be allowed to discharge into a chamber with a pressure  $p_a$  greater than  $p_2$ , the action will be very undesirable. In the first place, the pressure will fall nearly in the normal way till it reaches the chamber pressure  $p_a$ ; beyond the point at which this pressure is reached the pressure falls below  $p_a$  and then rises again above  $p_a$ , and beyond the nozzle the jet will be unstable. This action is called over-expansion. There is no reason why it should occur under normal conditions. If for any reason the designer has reason to fear that over-expansion may occur at an overload of the turbine he will try to avoid it by cutting his nozzles a little shorter than his design indicates, and accept the incomplete expansion which results.

**Design of a Nozzle.** — To illustrate the methods of this chapter a design will be worked out for a de Laval nozzle having the following conditions:

Gauge pressure at entrance . . . . .	138.2 pounds
Vacuum at exit . . . . .	25.9 inches
Weight of steam discharged per hour . . .	600 pounds
Taper of conical part of nozzle 1 in 6.	

Taking the pressure of the atmosphere at 14.7 pounds per square inch, the initial pressure becomes 152.9 pounds absolute. The conversion table on page 75 of the *Tables* gives 12.7 pounds as the equivalent of 25.9 inches of mercury; consequently the final pressure is 2 pounds absolute. The pressure at the throat may be taken as 0.58 of the initial pressure 152.9, so that the throat pressure is 88.7 pounds. Pages 81 et seq. of the temperature-entropy table give for entropy 1.56 the following properties:

$t_1 = 360^\circ$ ,	$p_1 = 152.9$ ,	$C_1 = 1187.1$ ,	
$t_2 = 319^\circ$ ,	$p_2 = 88.3$ ,	$C_2 = 1143.2$ ,	$v_2 = 4.75$ ,
$t_3 = 126^\circ$ ,	$p_3 = 1.99$ ,	$C_3 = 904.9$ .	

In entering the temperature-entropy table we will usually take the temperature of saturated steam corresponding most nearly to the given pressure and select the column of entropy which gives the quality or dryness factor nearest unity; or for

superheated steam that column which agrees nearest with the given superheat.

Since the rounded entrance to the throat of the nozzle is short no allowance will be made for friction up to that place. The available heat will therefore be

$$1187.1 - 1143.2 = 43.9 \text{ B.T.U.,}$$

and the velocity at the throat will be

$$V_t = 223.7 \sqrt{43.9} = 1484 \text{ feet per second.}$$

The area in square feet is given by equation (6), page 16, to discharge  $w$  pounds per second. The weight per second in this case is  $600 \div 3600$ , and the factor 144 may be introduced to reduce the area to square inches so that the area at the throat is

$$a_t = \frac{144 \times 600 \times 4.75}{3600 \times 1484} = 0.077 \text{ square inches.}$$

The value of  $\gamma$  for the entire nozzle may be taken as equal to 0.10, consequently the available heat for producing the exit velocity is

$$0.9 (1187.1 - 904.9) = 0.9 \times 282.2 = 254.0 \text{ B.T.U.,}$$

and the exit velocity is

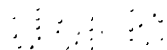
$$V_e = 223.7 \sqrt{254.0} = 3566 \text{ feet per second.}$$

In order to find the specific volume at exit we have the fact that 254.0 B.T.U. were changed into kinetic energy, so that of the initial heat contents 1187.1 there will remain in the steam

$$1187.1 - 254.0 = 933.1 \text{ B.T.U.}$$

at the temperature  $126^\circ$ . We can find the corresponding specific volume by interpolating in the temperature-entropy table between the entropies 1.60 and 1.61 as follows:

Heat contents.		Volumes.	
934.2	933.1	143.0	142.0
928.3	928.3	142.0	0.8
<hr/>		<hr/>	
5.9	: 4.8	:: 1.0	: 0.8. Therefore 142.8 = $v_8$ .



The exit area is therefore

$$a_e = \frac{144 \times 600 \times 142.8 \times 142.8}{3600 \times 3566} = 0.964.$$

The diameters at the throat and at the exit are therefore taken as

$$d_t = 1\frac{5}{8} \text{ inch, and } d_e = 1\frac{1}{8} \text{ inches.}$$

The increase in diameter is

$$1\frac{1}{8} - 1\frac{5}{8} = \frac{3}{4}$$

of an inch; since the taper is one in six the length of the nozzle from throat to exit will be

$$\frac{3}{4} \times 6 = 4\frac{1}{2} \text{ inches;}$$

allowing half an inch for entrance and throat will give 5 inches for the length of the nozzle.

## CHAPTER III

### JETS AND VANES

THE fundamental principles of turbines are the same whether they are driven by water or by steam; but the use of an elastic fluid like steam instead of a fluid like water, which has practically a constant density, leads to differences in the application of those principles. One feature is immediately evident from the problem on page 25, i.e., that exceedingly high velocities are liable to be developed. In that problem it is computed that steam flowing from a pressure of 138.2 pounds by the gauge into a vacuum of 26 inches of mercury will develop a velocity of 3566 feet per second. The absolute pressures are 152.9 and 2.0 pounds, so that the difference of pressure is nearly 151 pounds. This range of pressure corresponds to a head of

$$151 \times 144 \div 62.4 = 348 \text{ feet,}$$

and such a head will give a velocity of

$$V = \sqrt{2 \times 32.16 \times 348} = 150 \text{ feet per second;}$$

that is the steam velocity is more than twenty times that of the water. But so great an hydraulic head, or fall of water, is seldom if ever applied to a single turbine, and would be considered to be inconvenient. One hundred feet is a large hydraulic head yielding a velocity of 80 feet per second, and twenty-five feet, yielding a velocity of 40 feet per second, is considered an effective head. If heads of 300 feet and upward were frequent, it is likely that compound turbines would be developed to use them; except for small powers, steam-turbines are always compound, that is the steam flows through a succession of turbines, which may therefore run at more manageable speeds.

The great velocities that are developed in steam-turbines, even when compounded, require careful reduction of clearances, and although they are restricted to fractions of an inch the question of leakage is very important. Another feature in which steam-turbines differ from hydraulic turbines is that steam is an elastic fluid which tends to fill any space to which it is admitted. The influence of this feature will appear in the distinction between impulse and reaction turbines.

**Impulse.** — If a well-formed stream of water flows from a conical nozzle on a flat plate it spreads over it smoothly in all directions and exerts a steady force on it. If the velocity of the stream is  $V_1$  feet per second, and if  $w$  pounds of water are discharged per second, the force will be very nearly equal to



FIG. 5.

$$P = \frac{w}{g} V_1. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This is the force exerted by the water on the plate; it is directed toward the right and has a positive sign following the usual convention. The force  $P$  is equal and opposite to the force exerted by the plate on the water, which force can be computed from the retardation or negative acceleration of the water. The velocity is changed by the action of the plate from  $V_1$  feet per second toward the right, to zero; the retardation is therefore  $V_1$  feet per second, which is to be multiplied by the mass to find the retarding force. The retarding force is directed toward the left, and would have a negative sign if expressed analytically. Here  $g$  is the acceleration due to gravity and is equal to 32.16 feet per second.

A force exerted by a jet or stream of fluid on a plate or a vane is called an *impulse*. It is important to keep clearly in mind that we are dealing with velocity, with change of velocity or acceleration, and with force, and that the force is measured in the usual way. The use of a special name for the force which is developed in this way is unfortunate, but it is too well established to be neglected.

**Flat Vane.** — If the plate or vane moves with the velocity of  $V$  feet per second in the direction of the jet, the change in velocity or negative acceleration will be  $V_1 - V$  feet per second, and the force or impulse exerted on the plate will be

$$P = \frac{w}{g}(V_1 - V).$$

This force in one second will move the distance  $V$  feet and will do the work

$$\frac{w}{g}(V_1 - V)V = \frac{w}{g}(V_1V - V^2) \text{ foot-pounds.} \quad (2)$$

Since the vane would soon move beyond the range of the jet, it would be necessary, in order to obtain continuous action on a motor, to provide a succession of vanes, which might be mounted on the rim of a wheel. There would be, in consequence, waste of energy due to the motion of vanes in a circle and to splattering and other imperfect action. Under favorable conditions the waste from these causes may be as little as 10 per cent for a water wheel. But if conditions are unfavorable, and especially if the velocity of the jet were high, the water would not spread smoothly over the plate or vane, and the motor would show a poorer efficiency.

Now steam has an exceedingly high velocity when discharged from a nozzle, and the jet is more easily broken, so that adverse influences have even a worse effect than on water, and there is the greater reason for following methods which tend to avoid waste. Also the nozzle must be so formed as to expand the steam down to the back-pressure, or expansion will continue beyond the nozzle with further acceleration of the steam under unfavorable conditions.

It is easy to show that the best efficiency of the simple action of a jet on a vane will be obtained by making the velocity  $V$  of the vane half the velocity  $V_1$  of the jet. In equation (2) the jet velocity  $V_1$  is constant, but the velocity  $V$  of the wheel may vary; to find the maximum work, differentiate with regard to the

variable  $V$  and equate the differential coefficient to zero, giving

$$\frac{d}{dV} \left[ \frac{w}{g} (V_1 V - V^2) \right] = 0. \quad \therefore V = \frac{1}{2} V_1.$$

This value when introduced into equation (2) gives for the work on the vane

$$\frac{1}{4} \frac{w}{g} V_1^2,$$

but the kinetic energy of the jet is

$$\frac{1}{2} \frac{w}{g} V_1^2,$$

so that the efficiency is

$$\frac{1}{4} \frac{w}{g} V_1^2 \div \frac{1}{2} \frac{w}{g} V_1^2 = 0.5.$$

This efficiency is to be affected by a factor to allow for imperfect action of the jet and for friction of the wheel on its axis; for a flat vane wheel the factor may be as small as 0.7, so that the real efficiency may be 0.35 for a water wheel. For steam the efficiency may be lower.

**Cylindrical Vane.** — If the flat vane in Fig. 5 be replaced by a semi-cylindrical vane, as in Fig. 6, the direction of the stream will be reversed and the impulse will be twice as great. If the vane has the velocity  $V$  in the direction of the jet, the relative velocity of the jet with regard to the vane will be



FIG. 6.

$$V_1 - V,$$

and neglecting friction this velocity may be attributed to the water where it leaves the vane. This relative velocity at exit will be directed toward the rear, so that the absolute velocity will be

$$V - (V_1 - V) = 2V - V_1.$$

The change of velocity or retardation will be

$$V_1 - (2V - V_1) = 2(V_1 - V),$$

and the impulse is consequently

$$P = \frac{w}{g} \cdot 2 (V_1 - V).$$

The work of impulse becomes

$$\frac{w}{g} \cdot 2 (V_1 - V) V = 2 \frac{w}{g} (V_1 V - V^2). \quad \dots (3)$$

The maximum occurs when

$$\frac{d}{dV} (V_1 V - V^2) = V_1 - 2V = 0, \quad \text{or} \quad V = \frac{1}{2} V_1.$$

But this value when introduced into the expression (3) gives for the work done on the vane

$$\frac{1}{2} \frac{w}{g} V_1^2,$$

which is equal to the kinetic energy of the jet, so that the efficiency without allowing for losses appears to be unity. A water wheel made on virtually this same principle may show a real efficiency of 0.90.

**Reaction.** — If a stream of water flows through a conical nozzle into the air with a velocity  $V_1$  a force

$$R = \frac{w}{g} V_1^2 \quad \dots \dots \dots (4)$$

will be exerted tending to move the vessel from which the flow takes place in the contrary direction. Here again  $w$  is the weight discharged per second, and  $g$  is the acceleration due to gravity in feet per second. The force  $R$  is called the *reaction*, a name that is so commonly used that it must be accepted. Since the fluid in the chamber is at rest, the velocity  $V_1$  is that imparted by the pressure in one second, and is therefore an acceleration, and the force as written in equation (4) is measured by the product of the mass and the acceleration. However elementary this may appear, it should be borne carefully in mind to avoid future confusion.

If steam is discharged from a proper expanding nozzle, which reduces the pressure to that of the atmosphere, its reaction will be very nearly represented by equation (4); but if the expansion is incomplete in the nozzle it will continue beyond, and the added acceleration will affect the reaction. The reaction in this case will be somewhat less than for complete expansion in the nozzle.

The reaction of a steam jet from a nozzle has found some favor as a means of investigating the experimental conditions of flow of steam. When the expansion is complete the results of experiments are consistent with those from other experimental methods, but when the expansion is incomplete results are difficult of interpretation and are inconsistent among themselves.

The velocity of the jet depends on the pressure in the chamber, and, if it can be maintained, the velocity will be the same relatively to the chamber when the latter is allowed to move. The work will in such case be equal to the product of the reaction (computed by equation (4)) and the velocity of the chamber. There is no simple way of supplying fluid to a chamber which moves in a straight line, and a reaction wheel supplied with fluid at the center and discharging through nozzles at the circumference is affected by centrifugal force. As there is no example at present of a pure reaction steam-turbine, it is not profitable to go further in this matter. It is, however, important to remember that velocity, or increase of velocity, is due to pressure in the space or chamber under consideration, and that such velocity is relative to that chamber. This conception will be used to distinguish between impulse steam-turbines and impulse and reaction turbines.

## CHAPTER IV

### SIMPLE IMPULSE TURBINES

THE steam-turbine is a form of steam-engine, and it is convenient to base its general theory on the same principles as for reciprocating engines.

The second law of thermodynamics indicates that the maximum efficiency for any heat engine working between the temperature  $t_1$  and  $t_2$  is

$$e_c = \frac{T_1 - T_2}{T} \quad \dots \quad (1)$$

This is the efficiency of an engine working on a reversible cycle, such as Carnot's cycle. A discussion of that cycle and its significance will be found in any work on thermodynamics. Though there is interest in comparing the actual efficiency of any engine (for example a steam-turbine) with the maximum efficiency, it does not form a desirable basis of comparison of steam-engine performance.

**Rankine's Cycle.** — An important investigation for the steam-engine may be made by aid of the accompanying figure, which represents the indicator diagram from a steam-engine without clearance and with a non-conducting cylinder. Steam is admitted at an absolute pressure  $p_1$  from  $a$  to  $b$ ; adiabatic expansion follows from  $b$  to  $c$ ; finally the steam is exhausted from  $c$  to  $d$  at the pressure  $p_2$ . The external work during admission for one pound of steam having the quality  $x_1$  is

$$p_1 v_1 = p_1 (x_1 u_1 + \sigma);$$

the external work during expansion is

$$E_1 - E_2 = \frac{1}{A} (q_1 + x_1 p_1 - q_2 - x_2 p_2);$$

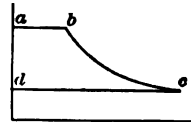


FIG. 7A.

and the external work of exhaust is

$$p_2 v_2 = p_2 (x_2 u_2 + \sigma),$$

which must be subtracted since it is done by the piston on the steam. The effective work of the cycle is

$$p_1 v_1 + E_1 - E_2 - p_2 v_2;$$

or substituting the proper values,

$$W = \frac{1}{A} (q_1 + x_1 p_1 + A p_1 x_1 u_1 - q_2 - x_2 p_2 - A p_2 x_2 u_2) + (p_1 - p_2) \sigma;$$

the last term is small and may be dropped.

Remembering that

$$r = \rho + A p u,$$

we have

$$W = \frac{1}{A} (q_1 + x_1 r_1 - q_2 - x_2 r_2). \quad . \quad . \quad . \quad (2)$$

The values of  $r_1$ ,  $r_2$ ,  $q_1$  and  $q_2$  may be taken from Tables I, II, or III of the *Tables*, so that when  $x_1$  has been assigned  $x_2$  may be computed by the equation

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2; \quad . \quad . \quad . \quad . \quad (3)$$

or the values of the heat contents

$$x_1 r_1 + q_1 \quad \text{and} \quad x_2 r_2 + q_2$$

may be taken directly from the temperature-entropy table.

By the first law of thermodynamics the difference between the heat supplied to an engine and the heat rejected is equivalent to the work done, provided that there are no losses; therefore

$$Q_1 - Q_2 = x_1 r_1 + q_1 - (x_2 r_2 + q_2). \quad . \quad . \quad . \quad (4)$$

This most important conclusion can be stated as follows: the heat changed into work by a steam-engine working on Rankine's cycle is equal to the difference in the heat contents of the steam supplied to and exhausted by the engine.

If the steam is superheated the heat contents at admission become approximately equal to

$$c_p (t_1 - t_s) + r_1 + q_1,$$

where  $c_p (t_1 - t_s)$  is the heat of superheat. The heat contents in the temperature-entropy table is, as already explained, obtained by a more precise way than that just written. But in any case the heat changed into work on Rankine's cycle may be obtained as stated above, whether the steam at admission or at exhaust is moist or superheated.

Now the heat required to raise one pound of water from  $t_2$  to  $t_1$  degrees and to partially vaporize it is

$$Q_1 = x_1 r_1 + q_1 - q_2, \quad . . . . . (5)$$

consequently the efficiency is

$$e = \frac{Q_1 - Q_2}{Q_1} = \frac{x_1 r_1 + q_1 - x_2 r_2 - q_2}{x_1 r_1 + q_1 - q_2} = 1 - \frac{C_2 - q_2}{C_1 - q_2}. \quad . . . (6)$$

If the steam is superheated the heat contents can of course be taken from the temperature-entropy table just as for moist steam.

**Application to Steam-turbines.** — The proposition just stated, i.e. that the heat changed into work on Rankine's cycle is equal to the difference of the heat contents, can be applied to steam-nozzles and to the general problem of steam-turbine design. On page 14 the customary deduction of velocity of flow through an orifice yielded a form identical with the right-hand member of equation (4) as the heat equivalent of the kinetic energy of a pound of steam at exit. As there stated, it may be considered that the available heat is applied to producing velocity in the steam instead of acting on a piston.

The conventional method of treating the general problem of steam-turbine design is to compare the actual steam consumption with the consumption of Rankine's cycle. The ratio of the latter under favorable conditions may vary from 0.6 to 0.7.

**Economy of Steam-engines.** — The performance of steam-engines is commonly stated in pounds of steam per horse-power per hour, and this custom is extended to steam-turbines. This

method does not take into account the effects of steam-pressure, vacuum, superheating, or steam-jacketing, and is liable to be misleading, especially for superheated steam or for use of steam-jackets on reciprocating engines. But if steam-turbines are supplied with moist steam, and if they are separated into condensing and non-condensing turbines, there is little objection to the method.

A method which is free from the objections against rating steam-engines in pounds per horse-power per hour is to base comparisons on thermal units, usually on thermal units per horse-power per minute. Now a horse-power is 33,000 foot-pounds per minute, consequently there will be

$$33,000 \div 778 = 42.42 \text{ B.T.U.}$$

transformed into work per horse-power per minute.

The steam consumption of an engine is determined by measuring the power by the indicator or otherwise, and by weighing the water supplied to the boiler or withdrawn from a condenser.

The steam consumption for Rankine's cycle can be computed from the consideration that

$$x_1 r_1 + q_1 - (x_2 r_2 + q_2) = C_1 - C_2$$

thermal units are changed into work per pound of steam. This method can of course be applied to superheated steam by aid of the temperature-entropy table. The steam required per horse-power per hour is

$$W_R = \frac{33,000 \times 60}{778 [x_1 r_1 + q_1 - (x_2 r_2 + q_2)]} = \frac{33,000 \times 60}{778 (C_1 - C_2)} = \frac{2545}{C_1 - C_2}; \quad (7)$$

the last form is applicable to both saturated or superheated steam.

As already stated, the heat required to heat water from  $t_2$  to  $t_1$  degrees and to vaporize it is

$$Q_1 = x_1 r_1 + q_1 - q_2 = C_1 - q_2,$$

and this holds in all cases, as it depends on the action of the boiler and not on the efficiency of the engine. If an engine uses

$W$  pounds of steam per horse-power per hour it will consume

$$\frac{(x_1 r_1 + q_1 - q_2) W}{60} = \frac{(C_1 - q_2) W}{60} \quad \dots \quad (8)$$

thermal units per horse-power per minute, the second form applying to both moist and superheated steam.

The efficiency of a steam-engine or steam-turbine can readily be determined after the thermal units per horse-power per minute consumed by the engine have been determined; it is sufficient to divide the number 42.42 by that quantity. This is an important item in steam-turbine design.

If this method is applied to Rankine's cycle making use of equations (7) and (8) the result is the same as given by equation (6), as of course it should be.

Conversely, if the efficiency of an engine is known the thermal units per horse-power per minute can be found by dividing 42.42 by the efficiency. And further, if the thermal units per horse-power per minute are known, the steam consumption can be found by multiplying by 60 and dividing by

$$x_1 r_1 + q_1 - q_2 = C_1 - q_2.$$

The second form applies equally to superheated steam, the heat contents being taken from the temperature-entropy table.

The discussion of the various features required for or convenient for turbine design must of course be considered before dealing with the design of a simple impulse turbine; they apply to all turbine design, and if not perfectly familiar should be carefully mastered, as continual reference will be made to them.

**General Case of Impulse.** — Having stated the general thermodynamic conditions for use in turbine design, we will develop the general principles of impulse turbines for use here, and in dealing with all other forms of impulse turbine.

In Fig. 7 let  $ac$  represent the velocity  $V_1$  of a jet of fluid, and let  $V$  represent the velocity of a curved vane  $cc$ . Then the velocity of the jet relative to the vane is  $V_2$  equal to  $bc$ . This relative velocity is drawn tangent to the end of the vane, and

such tangential entrance of the jet onto the vane is considered desirable for best efficiency; but there is probably little if any

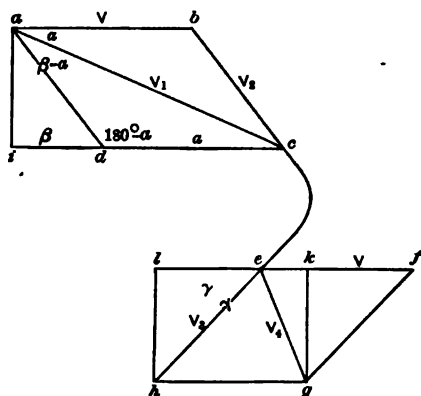


FIG. 7.

loss of efficiency if the relative velocity makes a small angle with the vane at entrance.

If it be supposed that the vane is bounded at the sides so that the stream cannot spread laterally, and if friction can be neglected, the relative velocity  $V_2$  at exit may be assumed to be equal to  $V_1$ . Its direction is along the tangent at the end of the

vane. The absolute velocity  $V_4$  can be found by drawing the parallelogram  $efgh$  with  $ef$  equal to  $V$ , the velocity of the vane.

The absolute entrance velocity  $V_1$  can be resolved into the components  $ai$  and  $ic$ , at right angles to and along the direction of motion of the water. The former may be called the velocity of flow,  $V_f$ , and the latter the velocity of whirl,  $V_w$ . In like manner the absolute exit velocity may be resolved into the components  $ek$  and  $kg$ , which may be called the exit velocity of whirl,  $V_w'$ , and the exit velocity of flow,  $V_f'$ .

The kinetic energy corresponding to the absolute exit velocity  $V_4$  is the lost or rejected energy of the combination of jet and vane, and for good efficiency should be made small. The exit velocity of whirl in general serves no good purpose and should be small to obtain good results.

The change in the velocity of whirl is the retardation (or negative acceleration) which determines the driving force or impulse; and the change in the velocity of flow in like manner produces an impulse at right angles to the motion of the vane, which in a turbine is felt as a thrust along the shaft.

Let the angle  $acd$  which the jet makes with the line of motion of the vane be represented by  $\alpha$ , and let  $\beta$  and  $\gamma$  represent the angles  $bcd$  and  $leh$  which the tangents at the entrance and exit of the vane make with the same line.

The driving impulse is in general equal to

$$P = \frac{w}{g} (V_w - V_w') \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and the axial thrust is in general

$$T = \frac{w}{g} (V_f - V_f'); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

this latter may be written

$$T = \frac{w}{g} (V_1 \sin \alpha - V_2 \sin \gamma) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

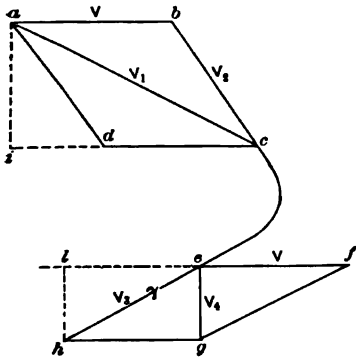


FIG. 8.

The conditions for no velocity of whirl at exit are represented by Fig. 8, in which the absolute velocity at exit  $V_4$  is vertical. In this case the impulse becomes

$$P = \frac{w}{g} V_1 \cos \alpha; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

and the work delivered to the vane per second is

$$W = PV = \frac{w}{g} V V_1 \cos \alpha \quad . \quad (13)$$

The kinetic energy of the jet is

$$\frac{w V_1^2}{2g},$$

consequently the efficiency is

$$e = 2 \frac{V}{V_1} \cos \alpha \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The relations of the angles may be found as follows from Fig. 8:

$$V_1 \sin \alpha = V_2 \sin \beta \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$V = V_2 \cos \gamma = V_2 \cos \gamma \quad . \quad . \quad . \quad . \quad . \quad (16)$$

because  $V_3$  is equal to  $V_2$ .

$$V = V_1 \cos \alpha - V_2 \cos \beta; \quad . \quad . \quad . \quad . \quad . \quad (17)$$

from which

$$V_1 \cos \alpha - V_1 \frac{\sin \alpha}{\sin \beta} \cos \beta = V_1 \frac{\sin \alpha \cos \gamma}{\sin \beta}.$$

$$\therefore \sin \beta \cos \alpha - \cos \beta \sin \alpha = \sin \alpha \cos \gamma \quad . \quad . \quad (18)$$

and

$$\sin (\beta - \alpha) = \sin \alpha \cos \gamma. \quad . \quad . \quad . \quad . \quad (19)$$

The equations just deduced may be applied to the computation of forces, work, and efficiency when  $w$  pounds of fluid per second are discharged from one or several nozzles and act on one or a number of vanes; that is, they are directly applicable to any simple impulse turbine not allowing for friction. The same remark applies to variants of this discussion presented later.

*Example.* Let  $V_1$ , the velocity of discharge, be 3566 feet per second as computed for a nozzle on page 25, and let  $\alpha = \gamma = 20^\circ$ . By equation (19)

$$\sin (\beta - \alpha) = \sin \alpha \cos \gamma = 0.3420 \times 0.9397 = 0.3214.$$

$$\therefore \beta - \alpha = 18^\circ 45'; \quad \beta = 38^\circ 45'$$

$$V_2 = V_1 \frac{\sin \alpha}{\sin \beta} = 3566 \frac{0.3420}{0.6259} = 1949$$

$$V = V_2 \cos \gamma = 1949 \times 0.9397 = 1831$$

$$e = 2 \frac{V}{V_1} \cos \alpha = 2 \times \frac{1831}{3566} \times 0.9397 = 0.9650.$$

**No Axial Thrust.** — Some builders of impulse steam-turbines attribute importance to avoiding axial thrust, which can be done by making the entrance and exit angles of the vanes equal, provided that friction and other resistances can be neglected.

To show this let Fig. 9 be drawn with  $\gamma = \beta$ , and with  $V_3 = V_2$ ; then

$$V_f = ai = V_1 \sin \alpha = V_2 \sin \beta = V_3 \sin \gamma = hl,$$

and since there is no axial retardation there will be no axial thrust.

The de Laval turbine has only one set of nozzles which expand the steam at once to the back-pressure, and consequently the velocity of the vanes is very high, and even with small wheels it is difficult to balance them satisfactorily. This difficulty is met by the use of a flexible shaft, and

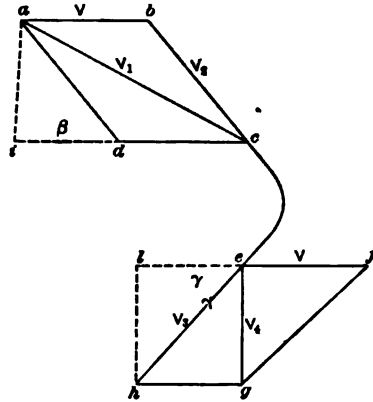


Fig. 9.

consequently axial thrust is likely to be troublesome; as a matter of fact the turbine is so arranged that the axial force shall be a pull. For most types of impulse turbines axial thrust, so long as it is not excessive, is not likely to give trouble, and can be taken care of by a small bearing or collar, which is needed to insure the adjustment of axial clearance.

Making  $\gamma = \beta$  in equation (18) gives

$$\sin \beta \cos \alpha - \cos \beta \sin \alpha = \sin \alpha \cos \beta.$$

$$\therefore \cot \beta = \frac{1}{2} \cot \alpha; \quad \dots \dots \dots (20)$$

and consequently from inspection of Fig. 9 it is evident that  $V$  is half the velocity or whirl, or

$$V = \frac{1}{2} V_1 \cos \alpha. \quad \dots \dots \dots (21)$$

If this value is carried into equations (13) and (14) the work and efficiency become

$$W = \frac{1}{2} \frac{w}{g} V_1^2 \cos^2 \alpha$$

$$e = \cos^2 \alpha. \quad \dots \dots \dots (22)$$

The following problem, as compared with that on page 40, shows that there is a gain of nine per cent from using the small exit angle.

*Example.* If, as in the preceding case, the velocity of discharge is 3566 feet per second, and if  $\alpha$  is  $20^\circ$ , we have the following results:

$$\cot \beta = \frac{1}{2} \cot \alpha = \frac{1}{2} \times 2.747 = 1.373 \quad \therefore \beta = 36^\circ$$

$$V = \frac{1}{2} V_1 \cos \alpha = \frac{1}{2} \times 3566 \times 0.9397 = 1676$$

$$e = \cos^2 20^\circ = 0.883.$$

**Effect of Friction.** — The direct effect of friction is to reduce the exit velocity from the vane; resistance due to striking the edges of the vanes, splattering, and other irregularities, will reduce velocity both at entering and leaving. The effect of friction and other resistances is two-fold: they reduce the efficiency of the turbine by changing kinetic energy into heat, and they reduce the velocity at which the best efficiency will be obtained. A more explicit discussion of the effect of blade friction will be found on page 75; for simple impulse wheels it does not appear worth while to treat this matter quantitatively. Small reductions from the speed of maximum efficiency have little effect.

The question as to what change shall be made in the exit angle (if any) on account of friction will depend on the relative importance attached to avoiding velocity of whirl and axial thrust. To avoid axial thrust the exit angle should be slightly increased, so that the exit velocity of flow may be equal to the entrance velocity of flow. But if it is desired to make the exit velocity of whirl zero, then  $\gamma$  should be slightly decreased. The de Laval turbine has the entrance and exit angles equal, and is run at a peripheral speed less than indicated by the theory to avoid excessive stresses in the wheel.

**Shaft Horse-power and Turbine Horse-power.** — In the discussion of reciprocating engines it is customary to base efficiency and economy on indicated horse-power; afterwards allowance is made for the friction of the engines to determine the shaft

horse-power. This custom came because the indicated power is easily found, while the determination of the shaft horse-power may be troublesome or impossible. The thermodynamic discussion is properly based on the indicated power, because that is the power developed by the action of the steam on the piston.

There is no direct way of determining the power delivered by the steam to the blades of a steam-turbine, and consequently when the power is measured it must be the shaft horse-power. For marine steam-turbines the power is found by measuring the twist of the shaft; for turbines driving electric generators it may be inferred from the action of the generator. But in any case the thermodynamic discussion of the turbine must be based on the turbine horse-power delivered by the steam to its blades; from analogy this is sometimes called the indicated power. In some cases the frictional losses may be found for a turbo-generator by using the generator as a motor, deriving the current from outside sources; the turbine chamber should then have a good vacuum to reduce fluid friction; or the turbine wheels may be removed and the shaft properly loaded to have the same bearing pressure.

The mechanical efficiency of steam-engines varies from 0.85 to 0.93; the best results are from well-built engines under favorable conditions. The friction of steam-turbines is less and may be estimated from 10 per cent downward. This refers to the friction of bearings only; the fluid friction of the steam on vanes and the resistance to the rotation of the wheel in the steam which fills the turbine case are to be considered separately.

**Steam Friction.** — There are two ways of determining the influence of steam friction on the action of a turbine: (1) the friction for individual members of the turbine may be determined by experiments made for that purpose, similar to the experiments on nozzles; or (2) the total effect of friction on all the members may be estimated from a comparison of the actual economy of the turbine with the economy that might be expected were there no friction; afterward, the friction may be distributed

among the various members according to the discretion of the designer. Probably a combination of the two methods will give the best results.

The first method has been applied to nozzles like those for the de Laval turbine and the factors given on page 22 may be used with confidence. This method has also been applied to determining friction of disks rotating in steam. The friction of steam passing through the blades of the wheels, and also the resistance of blades, especially when not in full action, are best treated by the second method.

**Design of a Simple Impulse Turbine.** — The following computation may be taken to illustrate the method of applying the foregoing discussion to a simple impulse turbine of the de Laval type.

Conditions and data:

Brake horse-power . . . . .	140
Pressure by gauge in steam chest, pounds . . .	138.2
Pressure of atmosphere, pounds . . . . .	14.7
Vacuum inches of mercury . . . . .	26
Peripheral speed not to exceed, feet per second . .	1200
Revolutions per minute . . . . .	1200
Number of nozzles . . . . .	4
Angle of nozzles, degrees . . . . .	20
Friction factor for nozzles, $\gamma$ . . . . .	0.10
Efficiency for vanes . . . . .	0.90
Heat losses due to rotation, etc., factor . . .	0.90
Mechanical efficiency . . . . .	0.90
Blade angle at exit equal to entrance angle.	

The absolute pressure in the steam chest and the corresponding temperature are

$$p_1 = 138.2 + 14.7 = 152.9 \text{ pounds; } t_1 = 360^\circ.$$

At the vacuum of 26 inches the absolute pressure and the temperature are

$$p_2 = 2 \text{ pounds; } t_2 = 126^\circ.$$

The nozzle computation has already been given on page 24, but will be summarized to complete the problem.

The entropy table gives for entropy 1.56 and  $360^\circ$ , heat contents 1187.1; at temperature  $126^\circ$  the heat contents are 904.9. The available heat is

$$1187.1 - 904.9 = 282.2 \text{ B.T.U.}$$

The throat pressure is

$$0.58 \times 152.9 = 88.7 \text{ pounds.}$$

At this pressure the nearest temperature is  $319^\circ$ , and at entropy 1.56 the heat-contents are 1143.2, so that the available heat is

$$1187.1 - 1143.2 = 43.9,$$

which gives for the throat velocity

$$\sqrt{2 \times 32.16 \times 778} \sqrt{43.9} = 223.7 \sqrt{43.9} = 1484$$

feet per second, neglecting friction at entrance. The velocity at exit is

$$223.7 \sqrt{282.2 \times .9} = 3566$$

feet per second, with the factor  $\gamma = 0.1$  as indicated.

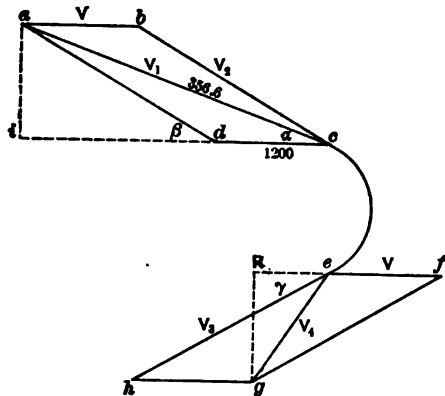


FIG. 10.

Fig. 10 is drawn with the angle  $\alpha$  equal to  $20^\circ$  and with the velocity from the nozzle 3566 feet per second, while the peripheral velocity is limited to 1200 feet per second. Com-



A method for determining the efficiency of the blades, taking account of friction, will be given in the next chapter; a crude method of multiplying the efficiency by a factor will be sufficient in this place; an arbitrary value (0.73) will be assigned.

The efficiency of Rankine's cycle between the temperatures of the steam-chest and exhaust is

$$e = 1 - \frac{C_2 - q_2}{C_1 - q_2} = 1 - \frac{904.9 - 94}{1187.1 - 94} = 0.258.$$

The factor to allow for friction in the nozzle was taken as  $\gamma = 0.1$ ; consequently 0.9 of the available heat was changed into kinetic energy of the jet. The factor to allow for disk friction and resistance was 0.1, so that only 0.9 of the work developed by the turbine was permanently applied to driving the shaft. There remains the mechanical efficiency of 0.9.

The final efficiency of the turbine is consequently estimated by the continuous product

$$\begin{array}{cccccc} \text{Nozzle;} & \text{Thermal;} & \text{Blades;} & \text{Disk;} & \text{Mech.} \\ 0.9 & \times & 0.258 & \times & 0.73 & \times & 0.9 & = & 0.138. \end{array}$$

It was shown on page 36 that one horse-power is equivalent to 42.42 B.T.U. per minute. The efficiency as just computed being 0.138, the turbine must be supplied with

$$42.42 \div 0.138 = 308 \text{ B.T.U. } \therefore$$

per shaft horse-power per minute.

There are required

$$x_1 r_1 + q_1 - q_2 = 1187.1 - 94.0 = 1093.1 \text{ B.T.U.}$$

to vaporize one pound of steam so that the steam per brake horse-power per hour is

$$308 \times 60 \div 1093.1 = 16.9 \text{ pounds.}$$

The total hourly consumption is

$$16.9 \times 140 = 2370 \text{ pounds.}$$

The design of a nozzle on page 24, which was made to conform to this case, gave for the throat and exit diameters

$$d_t = \frac{1}{8} \text{ inch} \quad d_e = \frac{1}{8} \text{ inch.}$$

For the sake of completeness the computation given there will be repeated here. In the first place it may be assumed that four nozzles will be used; there will therefore be nearly 600 pounds of steam per hour per nozzle, or

$$600 \div 3600 = \frac{1}{6} \text{ pound per second.}$$

The specific volume of the steam at the throat and at the exit from the nozzles will be 4.73 and 142.8 cubic feet. The areas in square inches will be

$$\text{Throat } a_t = \frac{144 \times 4.73}{6 \times 1484} = 0.077 \text{ square inch;}$$

$$\text{Exit } a_e = \frac{144 \times 142.8}{6 \times 3566} = 0.964 \text{ square inch;}$$

the corresponding diameters are as given above.

The blades may be made one thirty-second of an inch longer than the diameter of the nozzle at exit; this gives a blade length of  $1\frac{5}{8}$  inches.

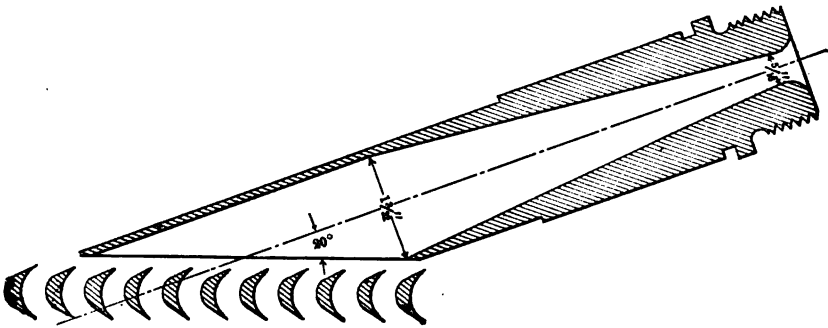


FIG. 12.

The section of a nozzle and some of the blades are shown by Fig. 12. The throat of the nozzle is made of uniform diameter ( $\frac{5}{16}$  inch) for about one-eighth of an inch to insure that the dimension shall be obtained and maintained; the nozzle is then flared to a ratio of one in six. The length of the conical part therefore becomes

$$6 \left( 1\frac{1}{8} - \frac{5}{16} \right) = 4\frac{5}{8} \text{ inches,}$$

which with the entrance and throat will give a length of five inches for the nozzle proper. The nozzle is continued by a cylindrical shell to the face of the wheel to avoid dissipation of the jet.

Before the blades can be laid out the diameter of the wheel and its other proportions must be determined. Since the peripheral speed is taken as 1200 feet per second, and there are to be 1200 revolutions per minute, the diameter of the pitch circle will be

$$1200 \div \frac{1200 \pi}{60} = 1.91 \text{ feet, or } 23 \text{ inches.}$$

For constructive reasons the axial width of the blades, as shown by Fig. 12, will be from three-eighths of an inch to half an inch for all sizes of turbines; in this case take half an inch. The pitch, or distance from blade to blade, may be about three-fourths of the axial width; in this case about three-eighths of an inch. The circumference of the pitch circle is 72.24 inches; if there are 180 blades the pitch will be

$$72.24 \div 180 = 0.402 \text{ inch.}$$

The nozzle in Fig. 12 is continued by a cylindrical shell which is cut off by a plane making an angle of  $20^\circ$  with the axis; the section of the shell is an ellipse having for its minor diameter  $1\frac{1}{8}$  inch, and for the major diameter

$$1\frac{1}{8} \div \sin 20^\circ = 1\frac{1}{8} \div 0.342 = 3\frac{1}{4} \text{ inches.}$$

As there are four nozzles the peripheral space occupied will be 13 inches. There will therefore be space for extra overload nozzles if desired, or another set of nozzles, expanding to the atmosphere only, may be provided for non-condensing action.

With the assigned dimensions for pitch and axial width, the blades can be laid out as shown by Fig. 12, the section being at half the blade length, that is at the pitch

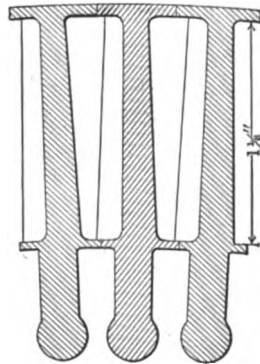


FIG. 13.

circumference of the wheel. The blades, which are of drop-forged steel, are inserted as shown by Fig. 13. They are slightly thicker at the root than at the tip, and the flanges at the tips form a continuous crown when assembled.

The section of the blade may be constructed as shown by Fig. 14. The angle  $\beta = 29^\circ 30'$  is laid off on each side of the center line and the sides are continued to give the proper axial width. The edges are made parallel to the center line for 0.02 of an inch to avoid a knife-edge. From the point  $a$  thus located a line  $ae$  is drawn perpendicular to the line  $de$ ; the

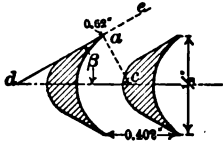


FIG. 14.

face is drawn from the center  $c$ , and is a circular arc with the proper angle  $\beta$  at entrance. The thickness of the blade should be sufficient to provide proper strength to resist centrifugal force and the impulse of the steam. A discussion of this matter for a similar case can be found on page 104. For our present purpose it will be sufficient to make the thickness three-eighths of the width at the root and one-fourth at the tip, giving

$$\begin{aligned} \text{Thickness, inner end, } \frac{1}{2} \times \frac{3}{8} &= \frac{3}{16} \\ \text{middle, } &\frac{5}{16}, \\ \text{outer end, } \frac{1}{2} \times \frac{1}{4} &= \frac{1}{8}. \end{aligned}$$

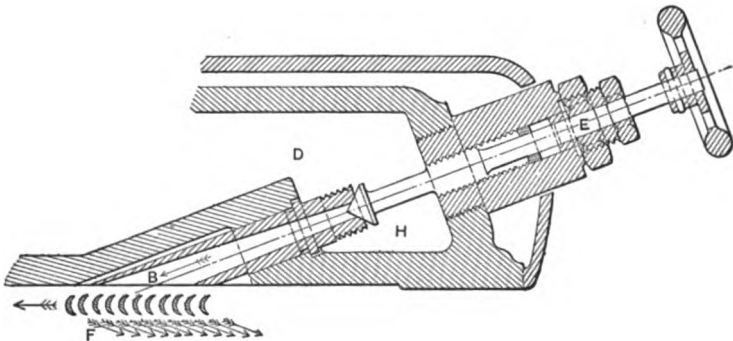


FIG. 15

A different construction for blade of impulse turbines is given on page 92, which provides a uniform passage for the flow of

steam. But for the de Laval turbine such considerations have less weight, as the passages do not run full of steam.

The method of setting the nozzles in the steam chest is shown by Fig. 15, together with the valve for controlling the nozzle. The end of the nozzle is threaded, to provide a ready means of withdrawing it when necessary.

Fig. 16 shows a section of the wheel which is forged solid from high-grade steel. To avoid piercing the wheel the shaft is flanged into sockets in the sides of the wheel.

The General Arrangement of a small de Laval turbine is shown by Fig. 17, in which *W* is the wheel carrying the blades on its perimeter; for small sizes the wheels are perforated for the shaft and are reënforced by massive hubs. Steam is supplied by an annular steam chest *cc*, in which are set the nozzles, as shown in Fig. 15; the steam from the wheel flows through the surrounding exhaust space and thence to the condenser.

Since the speed of rotation is very high it is necessary to gear down, as shown in the figure. On the turbine shaft there are cut right- and left-hand helical pinions meshing with gears on two parallel shafts as shown. The reduction is commonly in the ratio of ten to one.

A peculiarity of this turbine is the use of a flexible shaft which allows the wheel to rotate about an axis through its center of gravity at speeds above the critical speed; the outer end of the shaft has a spherical bearing to give greater freedom. The out-board ends of the gear shafts may be coupled to pumps, fans, or electric generators.

**Effect of Speed on Power.** — There is a marked difference between reciprocating engines and steam-turbines when run at other than the normal speeds. The power of a reciprocating

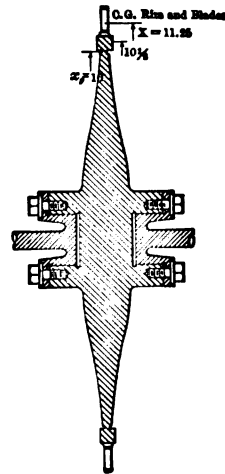


FIG. 16.

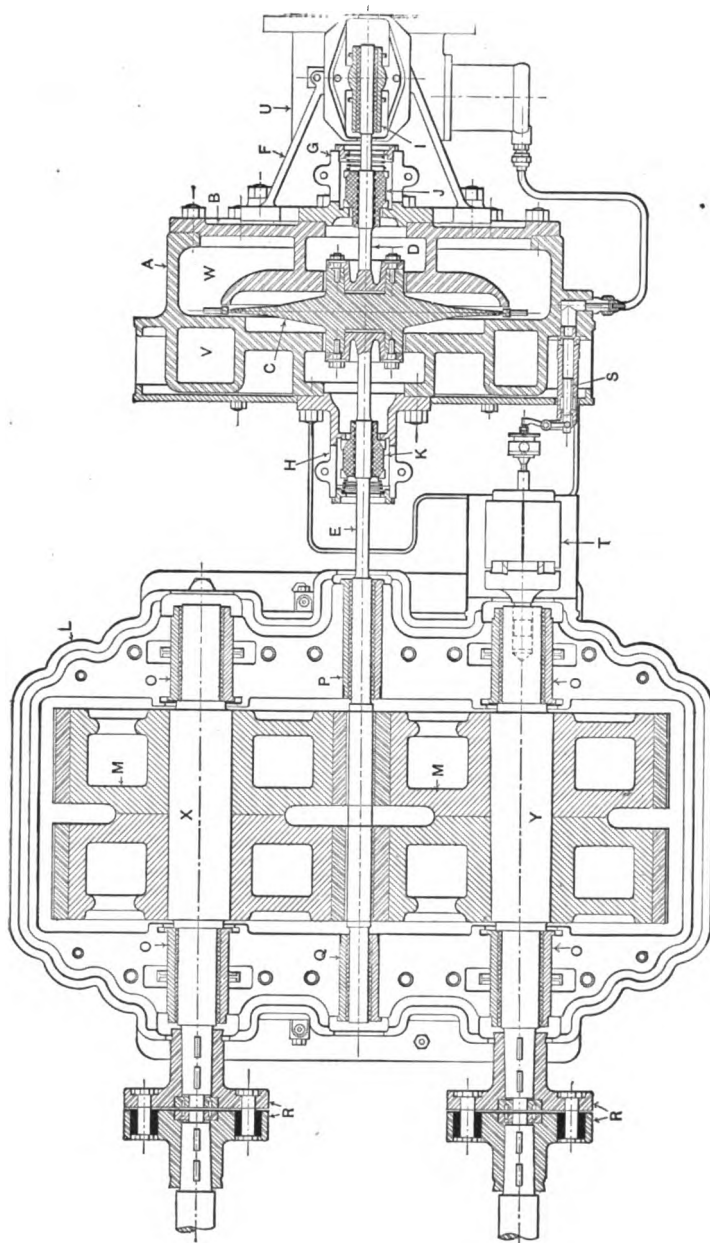


FIG. 17

engine varies directly with the speed; there is a slight reduction in mean-effective pressure with increase of speed, but the effect is insignificant. On the other hand, the power of a steam-turbine varies but little for comparatively large changes of speed. The flow of steam through the nozzles of an impulse turbine is sensibly constant for all conditions of running, provided that the drop of pressure is two-fifths of the initial pressure; for less drop of pressure the flow may be affected sensibly by changes of speed of the turbine, but not to a large extent. Reaction turbines cannot be dealt with so simply, but the general effect is similar. The power depends on the impulse and the speed of the blades; as the speed decreases the impulse increases, and so the power and efficiency change slowly.

The efficiency computed from the change of velocity in the preceding design was 0.81; had the velocity of the blades been half the initial velocity of whirl,

$$\frac{1}{2} \times 3351 = 1675,$$

instead of 1200, the efficiency would have been

$$\cos^2 20^\circ = 0.940^2 = 0.88,$$

and the horse-power would have been correspondingly greater. The loss of power on account of reduction of speed was therefore

$$(0.88 - 0.81) \div 0.88 = 0.08$$

for a reduction of nearly 30 per cent in speed. Tests on steam-turbines of various types confirm the statement that both power and efficiency decrease but slowly when a turbine is run at less than normal speed.

A similar condition holds for turbines which are run at more than the normal speeds, but as the normal speed is usually nearly as high as is safe, experiments at speeds much above the normal have not been made.

**Rotating Disks of Uniform Strength.** — The peripheral velocity of the disks of the de Laval wheels is limited by the resultant stresses in the disk, and consequently it is desirable to give a method of determining the stresses. A simple treatment of

this problem is possible provided that it is assumed (1) that the disk is subjected to centrifugal stresses only, (2) that the disk is solid, without a hole for the shaft, and (3) that the stresses are uniform throughout the disk. The treatment is approximate only, but appears to give satisfactory proportions in practice. Aside from any theoretical criticism of the method, it is clearly defective in that it does not take account of the torsion from the action of the steam on the buckets; if the buckets all receive steam the torsion may be uniform, but it will be irregular if the action of the steam is intermittent, as for the de Laval turbine.

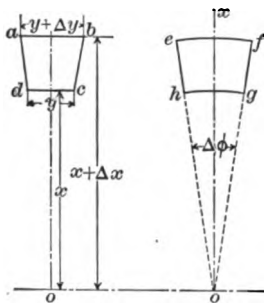


FIG. 18.

In addition to the preliminary assumptions it must be assumed that the disk is thin compared with the diameter. Fig. 18 shows, at the right, an elevation, and at the left a section of an elementary part of the disk. It is bounded by radial planes  $fg$  and  $he$ , and by cylindrical sections  $hg$  and  $ef$ . The thickness is assumed to increase from  $y$  to  $y + \Delta y$ , in order that the usual conventions concerning signs may apply; the disk actually increases in thickness toward the axis.

The volume of the element is approximately

$$y \cdot x \cdot \Delta\phi \cdot \Delta x.$$

The centrifugal force acting on it is

$$\frac{w}{g} xy \Delta x \Delta\phi \cdot x \alpha^2 = \frac{w}{g} x^2 y \alpha^2 \Delta x \Delta\phi, \quad . . . . (1)$$

when  $w$  is the weight per cubic inch of the material,  $g$  is the acceleration due to gravity, and  $\alpha$  is the angular velocity.

The element may be assumed to be subjected to the following uniform elastic forces:

Internal radial force,

$$- \sigma y x \Delta\phi. \quad . . . . . (2)$$

External radial force,

$$\sigma (y + \Delta y) (x + \Delta x) \Delta \phi;$$

or when differentials of higher orders are rejected,

$$\sigma (xy + x\Delta y + y\Delta x) \Delta \phi. \quad . \quad . \quad . \quad . \quad (3)$$

Tangential forces at each end  $eg$  and  $fh$ ,

$$\sigma y \Delta x.$$

The stresses are assumed to be constant and are represented by  $\sigma$ . The negative sign attached to the internal radial force signifies that it acts toward the axis. The tangential stresses may each be resolved into two components, one parallel to the radial line  $ox$  and the other perpendicular; those that are perpendicular to that line are opposed to each other and produce tangential equilibrium. Those that are parallel to  $ox$  provide a force

$$- 2 \sigma y \Delta x \cdot \frac{1}{2} \Delta \phi = - \sigma y \Delta x \Delta \phi, \quad . \quad . \quad . \quad . \quad (4)$$

which acts with the radial forces deduced above.

For equilibrium the radial forces when summed up must be equal to zero, therefore from (1), (2), (3), and (4) we have

$$\frac{w}{g} \alpha^2 x^2 y \Delta x - \sigma y x + \sigma x y + \sigma x \Delta y + \sigma y \Delta x - \sigma y \Delta x = 0;$$

or rearranging and passing to the limit,

$$\frac{dy}{y} = - \frac{w}{g} \frac{\alpha^2}{\sigma} x dx, \quad . \quad . \quad . \quad . \quad (5)$$

and integrating gives

$$\log_e y = - \frac{w}{2g} \frac{\alpha^2}{\sigma} x^2 + \text{const.} \quad . \quad . \quad . \quad . \quad (6)$$

At the center of the shaft where  $x_0 = 0$  the constant may be determined to be

$$\text{const.} = \log_e y_0$$

where  $y_0$  is the thickness of the disk at the axis. Substituting this value of the constant in equation (6) and solving for  $y$ ,

$$y = y_0 \frac{1}{e^{\frac{w \alpha^2 x^2}{2 g \sigma}}}. \quad . \quad . \quad . \quad . \quad (7)$$

This equation enables us to determine the contour of the section of the disk after the thickness at the axis is assigned and is in the form usually quoted.

In order to avoid the danger of rupture through the body of the disk which might throw off large fragments, it is customary to cut a groove just inside the rim, as shown in Fig. 16. If by any chance the turbine should speed up dangerously, the rim and attached parts outside this groove would be thrown off into the case, which is strong enough to stop the fragments; the turbine having lost its blades, in part if not altogether, would then slow down.

Consider that the rim and attached parts, including everything beyond the groove, do not contribute to the circumferential strength, but that the centrifugal force acting on them is distributed over the thinnest circumferential section of the groove. If the total weight of the rim and attached parts beyond the groove is  $W$  pounds, and if the center of gravity of a segment of those parts is at the radius  $X$  (see Fig. 16), then the total distributed centrifugal force is

$$\frac{W}{g} \alpha^2 X.$$

The area of the circumferential section at the groove where the thickness is  $y_g$  is

$$2 \pi x_g y_g,$$

and this area may be supposed to be subjected to a uniform stress  $\sigma_g$ . Consequently,

$$y_g = \frac{W \alpha^2 X}{2 \pi g \sigma_g x_g} \quad . . . . . (8)$$

gives the thickness at the groove. The thickness that the disk would have at the same place were it not grooved may be

$$y_1 = \frac{\sigma}{\sigma_g} y_g \quad . . . . . (9)$$

Having the thickness at one point, that at the center of the disk may be found by aid of equation (7); afterwards the thicknesses

at a sufficient number of points may be found by aid of equation (7) and the section may be drawn, as in Fig. 16.

*Example.* — Let it be assumed that the allowable stress at the groove of the wheel is 48,000 pounds per square inch, while the stress in the body of the wheel is 30,000 pounds. The blade length being  $1\frac{1}{8}$  inches, the half length will be nine-sixteenths of an inch; if then a sixteenth of an inch is allowed for the inner flange (Fig. 13) the radius of the edge of the disk will be

$$\frac{1}{2} \times 23 - \frac{5}{8} = 10\frac{7}{8} \text{ inches.}$$

If the section of the rim is made three-fourths of an inch square the radius to the middle will be  $10\frac{1}{2}$  inches. The perimeter will be 66.0 inches. Taking the weight of a cubic inch of steel at 0.28 of a pound, the weight of the rim, including the shanks of the blades, will be

$$66 \times (\frac{3}{4})^2 \times 0.28 = 10.4 \text{ pounds.}$$

If the weight of the extruding part of the blade is 0.35 of a pound then 180 blades will weigh 6.3 pounds. The total weight is consequently 16.7 pounds.

Equation (8), page 56, will give for the thickness of the disk at the groove

$$y_0 = \frac{W\alpha^2 X}{2\pi g\sigma x_0} = \frac{16.7 \times 400^2 \times \pi^2 \times 11.25}{2 \times \pi \times 12 \times 32.2 \times 48,000 \times 10},$$

where  $X = 11.25$  inches,  $x_0 = 10$  inches,  $g = 32.2 \times 12$ ,

and

$$\alpha = \frac{2\pi \times 12,000}{60} = 400\pi.$$

The thickness at the groove is therefore 0.255 of an inch, and the thickness at the inside of the rim is

$$0.255 \times \frac{4800}{3000} = 0.408 \text{ inch.}$$

In order to find the thickness of the disk at the axis, equation (7) may be transformed to give

$$y_0 = y_1 e^{\frac{w\alpha^2 x_1^2}{2g\sigma}}.$$

It is convenient to reduce the exponent by introducing the values which have already been used, giving

$$\frac{w\alpha^2 x_g^2}{2g\sigma} = \frac{0.28 \times 400^2 \times \pi^2 \times 10^2}{2 \times 32.2 \times 12 \times 30,000} = 1.92.$$

The base of the Napierian system of logarithms is

$$e = 2.71828,$$

and

$$y_0 = 0.408 e^{1.907} = 2.75 \text{ inches.}$$

The thickness can now be computed at various radii; for example, at 4 inches it is

$$y = \frac{2.75}{e^{\frac{w\alpha^2 x^2}{2g\sigma}}} = 2.02 \text{ inches.}$$

**Similar Wheels.** — If we consider two geometrically similar wheels, the larger having  $l$  times the diameter of the smaller, it is evident that Fig. 18 may represent the condition of an element of either wheel provided the scales are properly adjusted. Thus, if the larger wheel is four times the size of the other, we might let one inch represent four feet for the larger and one foot for the smaller. The volumes and masses for the two wheels will be proportional to the cube of the linear ratio, and this will apply also to the element.

The centrifugal force acting on an element will depend on the mass, the radius, and the square of the angular velocity; if the ratio of the angular velocities is  $a$  then the centrifugal forces will vary as

$$l^3 \cdot l \cdot a^2 = l^4 a^2. \quad \dots \dots \dots (1)$$

Considering the elastic forces acting on the element of Fig. 18 it is apparent that they vary as the area, and as the stress per square inch. If the stress is maintained constant then the forces will vary as the square of a linear dimension or as  $l^2$ .

Since both conditions must hold we have

$$l^4 a^2 \propto l^2,$$

$$\therefore a \propto \frac{1}{l}, \quad \dots \dots \dots (2)$$

that is the angular velocity or the number of revolutions must decrease proportionally to the increase of size. Thus, a wheel four times as large may have only one-fourth as many revolutions.

Or we may write

$$a\omega = \text{const.} = \frac{\alpha_1 R_1}{\alpha_2 R_2},$$

$$\therefore a_1 R_1 = a_2 R_2;$$

that is the linear velocity of the perimeter must be the same for similar wheels if the stress is to be the same.

On the other hand, if the weight of the blades and rim for a given wheel is increased the thicknesses must be increased proportionally, the character of the curves remaining the same.

These conclusions apply to wheels which have not uniform strength, provided that the conditions for the element of Fig. 18 are approximately correct.

The great importance of this discussion of similarity is that by its aid experiments on wheels of a type may be applied to all of that type of wheels.

## CHAPTER V

### PRESSURE COMPOUNDING

THE peripheral speed of simple impulse turbines is too high for general application, even with a reduction by the aid of gearing, and there are serious difficulties in the construction of turbines of that type to develop more than four or five hundred horse-power.

There are two methods of reducing the peripheral speed of impulse turbines, known as *pressure compounding* and *velocity compounding*; both methods may be used in conjunction.

A pressure compound turbine has a series of simple impulse turbines, as shown by Fig. 38, page 114, through which the steam passes in succession, much as is the case for a multiple-compound steam-engine; as with the steam-engine, one feature of the design is to arrange that there shall be the same amount of work developed in each stage, or that the work shall be distributed conveniently. It will appear that a large number of stages are required to give convenient peripheral speeds, and consequently a simple and reliable method of distribution of pressures is important. All the methods that have been proposed are based on adiabatic action, with modifications to allow for increase of entropy on account of degradation of heat through frictional and other disturbing influences.

**Effect of Friction on Entropy.** — It is well known that the influence of steam friction is to increase the entropy; the extent of this influence is readily shown by an example. As a fair mean of conditions for the design of a turbine intended to give good efficiency we may assume the following conditions: initial pressure, 150 pounds by the gauge, vacuum 28 inches of mercury; the corresponding absolute pressures are nearly 164.8 pounds and one pound absolute, for which the temperatures are  $366^{\circ}$  and  $102^{\circ}$ .

On page 81 of the *Steam and Entropy Tables* it appears that steam at  $366^{\circ}$  and entropy 1.56 is nearly dry, and has the heat contents 1193.3; at the same entropy and at temperature  $102^{\circ}$  the heat contents are 871.1, so that the available heat is 322.2 B.T.U.

Suppose that a turbine is to have two pressure stages, each having the same peripheral velocity of the blades, and that the heat changed into kinetic energy in the nozzles for each stage should consequently be the same. If the action were adiabatic the intermediate temperature and pressure could be found by dividing the available heat into equal parts, as shown in the following calculation:

Heat contents.	Entropy.	Temperatures.	Pressures.
1193.3	1.56	366	168.4
161.1			
1032.2	1.56	223.5	18.4
161.1			
871.1	1.56	102	1.0

The first and last lines are filled in at once from the *Entropy Table*; then half the available heat,

$$\frac{1}{2} \times 322.2 = 161.1,$$

is subtracted twice in succession. The heat contents at the intermediate temperature is found to be 1032.2 B.T.U., and interpolation in the table gives the corresponding temperature and pressure. The form here stated will be found convenient where there are a number of stages, especially as it gives a convenient check on the numerical work.

In turbine practice it is found that only a fraction of the adiabatically available heat can be changed into work and applied to driving the turbine. Suppose that the fraction in this case is 0.6, then

$$161.1 \times 0.6 = 96.7 \text{ B.T.U.}$$

will be changed into work in the first stage and the other,

$$161.1 - 96.7 = 64.4 \text{ B.T.U.},$$

will remain in the steam; in consequence the amount of moisture will be reduced and the *entropy will increase*. The conditions may readily be determined by interpolation on page 101 of the *Entropy Table*. Since only 96.7 B.T.U. have been taken from the steam there will remain as heat contents

$$1193.3 - 96.7 = 1096.6 \text{ B.T.U.},$$

which at  $223^{\circ}.5$  comes between the entropy column 1.65 and 1.66; to be precise, we may find by interpolation that the entropy becomes 1.651. This is the entropy at which the computation for the second stage should be made; but interpolation, and more especially cross interpolation, is tedious and unnecessary, because we may choose the nearest entropy column for our calculation; in this case 1.65. To emphasize this feature we will compute for the second stage in both 1.65 and 1.66 entropy columns.

Temperatures.	Pressures.	Heat contents.	
		Entropy 1.65	Entropy 1.66
223.5 102	18.4 1	1093.7	1100.6
		921.7	927.3
		172.0	173.3

The difference is about 0.75 of one per cent between the two columns, but since the nearer column will be chosen this reduces to 0.4 of one per cent, which can be neglected.

Taking the adiabatic available heat for the second stage as 172.0 (given above) there appears to be about seven per cent more than was found for the first stage. The velocities as computed for the two stages with  $\gamma = 0.10$  will be

$$V_1 = 223.7 \sqrt{0.9 \times 161.1} = 2695,$$

$$V_1' = 223.7 \sqrt{0.9 \times 172} = 2781.$$

A discrepancy of three or four per cent in steam velocity, as shown by this calculation, may not be a serious matter, but it can readily be avoided by any one of several ways that will be explained.

**Trial and Error.** — An obvious though tedious method is by trial and error. Take the mean of the adiabatic available heats just computed for the two stages

$$(161.1 + 172.0) \div 2 = 166.5.$$

Subtracting from the initial heat contents at entropy 1.56 we have

$$1193.3 - 166.5 = 1026.8,$$

which corresponds almost exactly to 219°. Assuming as before that 0.6 of the available adiabatic heat is changed into work we have for the heat contents in the steam approaching the second-stage nozzles,

$$1193.3 - 0.6 \times 166.5 = 1093.4,$$

which corresponds approximately to the heat contents at 219°, entropy 1.66.

The computation for the two stages can now be made as follows:

First Stage. Entropy 1.56.		Second Stage. Entropy 1.66.	
Temperature.	Heat contents.	Temperature.	Heat contents.
366	1193.3	219	1094.5
219	1026.7	102	927.3
	<hr/> 166.6		<hr/> 167.2

The agreement of these two results is close enough; in any case where it seems worth while a second approximation by the same method can be made. Other intermediate points could be found halfway between those given above if four stages are to be provided for, etc., and equal or unequal stages could be controlled with somewhat more complexity.

**Graphical Method.** — The first systematic method of finding intermediate pressures for pressure compound turbines appears to be a graphical method for which Mollier's diagram is convenient.

In Fig. 19 a portion of Mollier's diagram is repeated to a small scale. Through the intersection of the line of 165 pounds pressure with the saturation line, an isentropic line  $ad$  (1.56) is drawn down to the one-pound pressure line. Assuming as before that 0.6 of the adiabatic available heat is changed into work we divide the line  $ad$  into ten parts and take  $e$  at 0.6. Through  $e$

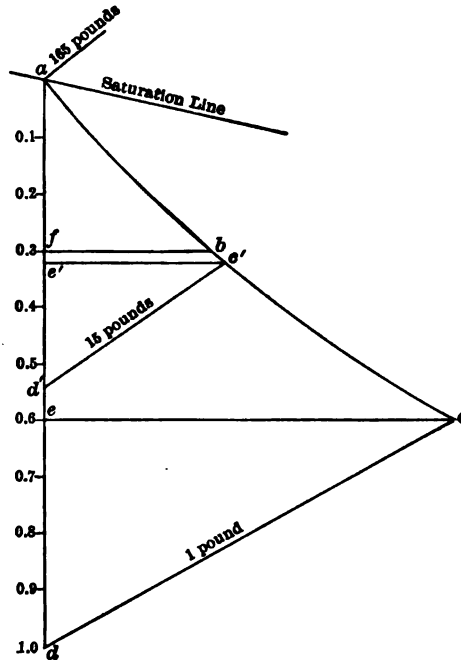


FIG. 19

draw the constant heat-contents line  $ec$ , to intersect the line  $dc$  at  $c$ ; this construction indicates the increase of entropy due to the remainder  $ed$  of the heat  $ad$ , which is not changed into work and therefore remains in the steam. The construction is repeated at  $ad'c'e'$  for the 15 pound line, thus locating another point  $c'$  of the curve  $ac'c$  which represents the true action of the steam on the assumption that the efficiency is constant, so that for each part of the entire operation 0.6 of the adiabatic available

heat is changed into work. Several points like  $c, c'$  should be constructed to locate the curve.

Assuming as before that there are two equal pressure stages, the line  $ae$ , which represents the heat changed into work, is bisected at  $f$  and the constant heat line  $fb$  is drawn to intersect the curve at  $b$ ; this indicates that the intermediate pressure is nearly 17 pounds. If there are several pressure stages the line  $ae$  may be divided into the corresponding number of portions, which may be equal or unequal as the design shall indicate.

**Direct Method.** — Taking advantage of the precision and consistency of the temperature-entropy table, the author has devised the following method for determining the intermediate pressures of a pressure-compound turbine.

An inspection of the temperature-entropy table or of a temperature-entropy diagram or Mollier's diagram, shows that the available heat between two given temperatures is greater at a greater entropy. Thus it appears that we have the following:

Entropy 1.56.		Entropy 1.65.	
Temperatures.	Heat Contents.	Temperatures.	Heat Contents.
223°.5	1032.2	223°.5	1093.7
102°.5	871.1	102°.5	921.7
	<hr/> 161.1		<hr/> 172.0

A further inspection of the tables shows that the heat contents at a certain entropy increase at nearly a uniform rate for considerable intervals of temperature, — for 20° intervals or even 40° intervals. We may therefore find the rate at which the available heat increases with the entropy by comparing the ratios

$$\frac{\Delta h}{\Delta t} \text{ and } \frac{\Delta h'}{\Delta t'};$$

where  $\Delta h$  is the increase of heat contents for the increase of temperature  $\Delta t$ . The comparison is best made at about the

middle temperature, between the temperature in the steam chest of the turbine and the temperature in the exhaust passage.

The application of the method is best shown by an example, taking the gauge pressure at entrance 150 pounds, and the vacuum at 28 inches of mercury; the corresponding temperatures are 366° and 102°. At entropy 1.56 and at these temperatures the heat-contents and the difference are

$$1193.3 - 871.1 = 322.2.$$

Half of the available heat is 161.1, which taken from the initial heat-contents gives

$$1193.3 - 161.1 = 1032.2,$$

which corresponds to 223°.5.

Assuming that 0.6 of the available heat is changed into work there will remain in the steam approaching the second set of nozzles the heat

$$1193.3 - 0.6 \times 161.1 = 1096.6.$$

This comes halfway between entropy 1.65 and 1.66, and we will take the trouble to interpolate; ordinarily it will suffice to take the nearest column. Taking the increment of temperature  $\Delta t$  as 40°, we have the following computation:

Temperature.	Entropy 1.56	Entropy 1.655.	Ratio.
	Heat Contents.	Heat Contents.	
243	1056.0	1122.8	...
203	1006.7	1069.6	...
	<hr/> 49.3	<hr/> 53.2	1.08

Since the last figure of the heat contents is subject to an error of computation amounting to half a unit, the accumulation of errors in the computation of the ratio is liable to be two or three units in the last place; therefore the ratio need not be stated nearer than half a per cent. It is of course immaterial at this place whether the temperature for which the computation is

made is  $223^\circ$  or  $224^\circ$ ; in any case the temperature will be taken without a fraction.

Now draw a diagram like Fig. 20, making  $h_1h_2$  represent the adiabatic available heat, and take a convenient scale for the ratio just computed and lay it off along the line from  $h_1$  to 1.08. Bisect the line  $h_1h_2$  at  $o$  and draw the inclined line  $ab$  cutting the line  $h_2b$  at 0.92. The heat  $h_1h_2$  may be divided into convenient portions for the design; in the type problem with two stages there will be two equal portions. At the middle of each portion draw an abscissa; it will indicate the factor for that portion, as will be explained later. With two equal stages Fig. 20 gives a factor 1.040 for the first stage and 0.960 for the second. The diagram is so simple and the results so easily obtained that we may commonly draw a freehand diagram and compute the factors.

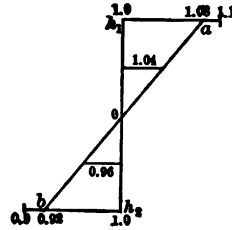


FIG. 20.

In applying the diagram, Fig. 20, the factors should be determined to the third decimal place in order that intermediate pressures may be distributed regularly; this instruction is not inconsistent with that for computing the ratio which is concerned with the determination of the middle temperature and that cannot and need not be determined more accurately. But there should not be much irregularity in the sequence of pressures of the several stages, which may be numerous; and for this purpose the factors must be taken with some precision.

	Heat portion.	Factor.	Heat contents, entropy 1.56.	Temperature.	Pressure.
First stage	$161.1 \times 1.040 =$		1193.3 167.5	$366^\circ.0$	164.8
Second stage	$161.1 \times 0.960 =$		1025.8 154.7	$218^\circ.5$	16.7
			871.1	$102^\circ.0$	1.0

The available adiabatic heat for our problems was determined to be 322.2 B.T.U., which gives 161.1 B.T.U. per stage. The portion per stage is to be multiplied by the proper factor as determined above, and subtracted in succession as on page 67.

**Ratios for Pressure-Distribution.** — The ratios for modifying the adiabatic method of distributing temperatures and pressures are corrective factors which vary but slowly, and commonly may be selected or interpolated from the following table:

RATIO FOR PRESSURE-DISTRIBUTION

Heat factor.	150 gauge pressure to 28 in. vacuum.	150 gauge pressure to atmosphere.	Atmosphere to 28 in. vacuum.
0.55	1.09	1.050	1.040
0.60	1.08	1.045	1.035
0.65	1.07	1.035	1.030
0.70	1.06	1.030	1.025
0.75	1.05	1.025	1.020

To illustrate the insensitiveness of this method of pressure distribution there has been calculated the following determinations of the mid-point for the typical case chosen for forming the table with three heat factors as follows:

Heat factor . . . . .	0.55	0.65	0.75
Intermediate temperature . . . . .	218	219	220
Pressure absolute . . . . .	16.5	16.9	17.2

Also as a further illustration of the same feature there is given in the following table the results of the determination of the mid-temperature and pressure for a range from 400° to 102°.

	Tabular value.	Special calculation.
Initial temperature . . . . .	366	400
Pressure by gauge. . . . .	150	232
Final temperature. . . . .	102	102
Vacuum . . . . .	28	28
Ratio . . . . .	1.07	1.075
Intermediate temperature . . . . .	231.8	232.1
Pressure absolute . . . . .	21.49	21.61

**Check Calculation.** — Whatever method of pressure distribution may be used in any case, the test of the method must be a computation of the available heat per stage.

For a two-stage turbine with gauge pressure of 150 pounds and a vacuum of 28 inches, the intermediate temperature was determined on page 67 to be  $218^{\circ}.5$  and the pressure 16.7 absolute. In the computation leading to this result it was assumed that

$$161.1 \times 0.6 = 96.7 \text{ B.T.U.}$$

were changed into work in the first stage, and that the steam approaching the second nozzles had the heat contents 1096.6 B.T.U. At the mid-temperature of  $218^{\circ}.5$  this figure is found at entropy 1.665. The available heat from  $218^{\circ}.5$  to  $102^{\circ}$  at this entropy is

$$1097.2 - 930.1 = 167.4,$$

instead of 167.5 assigned to the first stage.

In this case interpolation to half degrees and to 0.005 of a unit of entropy has been taken to give a little greater precision.

**Application to Six-stage Turbine.** — To further illustrate the direct method of determining intermediate pressures and temperatures the following application is made to a turbine with six equal stages, the pressure varying from 150-pounds gauge to 28-inch vacuum. The factors may be read from Fig. 21, which was constructed for a heat factor 0.65; the ratio 1.065 used in drawing that figure was determined by a process of somewhat greater refinement than that given on page 65, but it is not certain that it is more correct. The heat contents at  $366^{\circ}$  is 1193.3 and  $871.1$  at  $102^{\circ}$ , so that the mean assignment per stage is

$$(1193.3 - 871.1) \div 6 = 53.7.$$

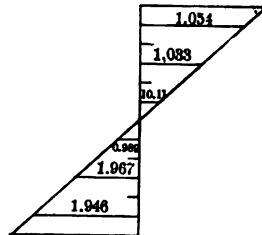


FIG. 21.

The heat assignments at the entropy 1.56 are:

$$53.7 \times 1.054 = 56.6$$

$$53.7 \times 1.032 = 55.4$$

$$53.7 \times 1.011 = 54.3$$

$$53.7 \times 0.989 = 53.1$$

$$53.7 \times 0.968 = 52.0$$

$$53.7 \times 0.946 = 50.8$$

$$\underline{\quad\quad\quad} 322.2.$$

The computation of the intermediate temperatures and pressures, and the check computation are given in the following table:

Heat contents, entropy 1.56. 1	Tempera- ture. 2	Pressure abso- lute. 3	Heat contents factor 0.65. 4	Entropy. 5	Heat drop. 6
1193.3 56.6	366.0 ...	164.8 ...	1193.3 34.9	1.560 ...	1193.3 1136.7 <u>56.6</u>
1136.7 55.4	313.0 ...	81.1 ...	1158.4 34.9	1.588 ...	1158.4 1101.5 <u>56.9</u>
1081.3 54.3	264.3 ...	38.1 ...	1123.5 34.9	1.618 ...	1123.5 1066.4 <u>57.1</u>
1027.0 53.1	219.3 ...	16.9 ...	1088.6 34.9	1.651 ...	1088.6 1031.9 <u>56.7</u>
973.9 52.0	177.4 ...	7.09 ...	1053.7 34.9	1.685 ...	1053.7 996.7 <u>57.0</u>
921.9 50.8	138.3 ...	2.76 ...	1018.8 34.9	1.722 ...	1018.8 962.1 <u>56.7</u>
871.1	102.0	1.00	983.9	1.761	...

In the check calculation the heat changed into work per stage is taken to be

$$53.7 \times 0.65 = 34.9.$$

This quantity, subtracted successively, gives the heat contents of the steam approaching the several sets of nozzles. The corresponding entropies are found by interpolating in the entropy table. At these entropies the heat contents are found for the initial and final temperatures for the several stages; the differences are the available heats for adiabatic action in these stages. The variation of the heat drops from the assigned value 56.6 is a measure of the precision of the method.

In this sample computation the work is carried to the limit of precision of the entropy table, in order to determine the degree of precision possible for the method. Usually it will be sufficient to take the nearest temperature of the table, except when interpolation gives half a degree; in like manner we may use the nearest entropy column, except when the entropy falls halfway between two columns. Experience shows that we then get just about the same degree of apparent precision as if the interpolation were carried to the limit. When this approximate computation is made the initial heat contents of the last column are liable to differ from those in column 4; thus, had we taken the nearest temperature and entropy from the table for the second stage we should have found

Temperature initial	313,	entropy	1.59,	heat contents	1159.8
	final 264	"	"		1102.6
			heat drop		57.2

Our knowledge of the properties of steam is not more precise than is represented by a single thermal unit; it is convenient to have tenths of units in our steam and entropy tables, and such a degree of concordance is required for the direct method of finding intermediate temperatures and pressures. But if that method shows discrepancies of half a thermal unit we get what we should expect. This discrepancy is likely to be the same for all cases, so that it is well to make computations at intervals, when a turbine has many stages, and to derive intermediate quantities by interpolation on a curve or otherwise.

**Unequal Stages.** — In the examples given for finding intermediate pressures the stages have had equal heat assignments and the line  $h_1h_2$  of Fig. 21 is divided into equal parts. In some cases the heats per stage are unequal, and then  $h_1h_2$  must be correspondingly divided; examples will be given later.

**Heat Factors; Overall and per Stage.** — The heat factor used in the method for determining the pressures and temperatures for a pressure compound turbine is the ratio of the steam per horse-power per hour for Rankine's cycle and the steam per *turbine* horse-power per hour. We can use the thermal units per horse-power per minute, instead of steam per horse-power per hour; or we may take the ratio of the actual turbine efficiency to the efficiency for Rankine's cycle.

A steam turbine of the type under consideration may have an actual steam consumption of 14.6 pounds per brake horse-power per hour working between the gauge pressure 150 pounds and a vacuum of 28 inches. The corresponding temperatures are  $366^\circ$  and  $102^\circ$ , which give for the available heat of Rankine's cycle

$$1193.3 - 871.1 = 322.2;$$

the steam per horse-power per hour for Rankine's cycle by equation (7), page 36, is

$$W_R = \frac{33,000 \times 60}{778 \times 322.2} = 7.9 \text{ pounds.}$$

Allowing a factor 0.9 for leakage and radiation and mechanical friction the steam per turbine horse-power will be

$$14.6 \times 0.9 = 13.14 \text{ pounds;}$$

consequently the *overall heat factor* is

$$7.9 \div 13.14 = 0.60.$$

When we come to analyzing the heat losses in a stage of a pressure compound turbine another heat factor is required. This is evident from the fact that the determination of the intermediate pressure for a two-stage turbine under the assumed conditions is made by assigning 167.5 B.T.U. (page 67) for the

$$h = .00052 \left( \frac{h-1}{h} \right) \frac{1}{(1-E)} = 1.14$$

EFFICIENCY AND HEAT-DEGRADATION 73

adiabatic action of the first stage, while the heat assumed to be changed into work in that stage is

$$161.1 \times 0.6 = 96.7 \text{ B.T.U.}$$

Consequently the ratio of the heat actually changed into work in the first stage to the adiabatically available heat is

$$96.7 \div 167.5 = 0.577;$$

this may be called the *heat factor per stage*.

The heat factor per stage may also be obtained by dividing the overall heat factor by the factor for temperature distribution. In the case in hand

$$0.6 \div 1.040 = 0.577;$$

this comes directly from the previous determination of the quantity

$$167.5 = 161.1 \times 1.040.$$

The conception of the two factors (the overall heat factor and the heat factor per stage) and of their relation is of much importance in the discussion of pressure compound turbines.

**Efficiency and Heat-Degradation.** — Care should be taken to avoid confusion in thought at this place. The fact that the overall heat factor is larger than the heat factor per stage may appear to indicate that there is an advantage from compounding *per se*; as a matter of fact, compounding is forced upon us for turbines (just as it is for compound engines) by extraneous conditions, and those conditions are such that good efficiency under practical conditions is possible only by compounding.

If a reciprocating engine could be made of non-conducting material, with such generous passages that there should be no loss of pressure from cylinder to cylinder, then there would be neither gain nor loss from compounding. If there were loss of pressure with non-conducting material, compounding would always show a loss of efficiency. But as engines are made of conducting materials there is a gain from compounding when high pressure is used.

In like manner if adiabatic action were possible in a turbine without loss from friction, and without loss of kinetic energy in passing from stage to stage, there would be no advantage or disadvantage from compounding except as compounding gives more manageable speeds. For various reasons simple turbines are possible only for small powers and at those powers and with the conditions under which they run, simple turbines are less economical than compound turbines.

In both compound engines and in compound turbines the heat not changed into work in a given stage is available for work in the next stage, but at a lower temperature and therefore with less efficiency; there is a degradation of heat from compounding and a corresponding loss of efficiency. Though the comparison is crude, it has some similarity to passing steam from the high-pressure steam chest to the low-pressure cylinder.

The assumption of an increasing entropy for the calculations for the successive stages of a turbine is a proper consequence of the partial compensation which comes from the drying of the steam by the heat not changed into work. The temperature distribution factor throws back that compensation into the earlier stages also.

**Number of Stages.** — The number of stages required for a pressure compound turbine will depend on the desired peripheral speed, and this will depend in part on the service of the turbine and in part on the material of the blades and the manner of fixing them to the wheels. Blades made of extruded metal (composition) can be run as high as 500 feet per second; if made of manganese bronze they may run at 600 feet per second; for speeds above 600 feet blades must be made of steel. On the other hand, it may be desirable to limit the peripheral speed to 200 or 300 feet per second for certain purposes.

*Example.* — Let it be required to determine the number of stages proper for a pressure compound turbine having a gauge pressure of 150 pounds and a vacuum of 28 inches, if the peripheral speed is to be limited to 300 feet per second.

The temperature corresponding to the assigned pressure and

vacuum are  $366^\circ$  and  $102^\circ$  and the adiabatically available heat (entropy 1.56) is

$$1193.3 - 871.1 = 322.2 \text{ B.T.U.}$$

Neglecting friction in the nozzles will give for the exit velocity from a nozzle which expands through the entire range of pressure,

$$V = \sqrt{2 \times 32.16 \times 778 \sqrt{322.2}} = 223.7 \times 17.95 = 4016$$

feet per second. If the angle of the nozzle is  $20^\circ$  the velocity of whirl will be

$$4016 \cos 20^\circ = 4016 \times 0.9397 = 3774.$$

For maximum efficiency the peripheral velocity should be half the velocity of whirl, or 1887 feet per second. This is

$$1887 \div 300 = 6.29$$

times the desired peripheral velocity. The velocity is proportional to the square root of the available heat, and conversely the available heat must be made proportional to the square of the velocity; consequently to reduce the velocity to the required degree we must have

$$\overline{6.29}^2 = 40 -$$

stages. This will give

$$322.2 \div 40 = 8.055 \text{ B.T.U.}$$

per stage, which for adiabatic action will give a velocity of

$$223.7 \sqrt{8.055} = 223.7 \times 2.838 = 635.$$

The best peripheral velocity will therefore be

$$\frac{1}{2} \times 635 \cos 20^\circ = \frac{1}{2} \times 635 \times 0.9397 = 299$$

feet per second. If allowance is made for nozzle friction there will be a slight corresponding reduction from this last figure. It is shown on page 51 that an impulse turbine may be run considerably under the normal peripheral speed without much loss of efficiency, and by this device the number of stages for a pressure compound turbine may be appreciably reduced.

**Effect of Blade-friction.** — A general discussion of the effect of blade-friction was given on page 42 in connection with the

simple impulse turbine; now that an estimate of the effect must be made in order to determine a probable value for the heat factor per stage, it is necessary to go more into particulars.

The simple impulse turbine is habitually designed with equal entrance and exit angles, and there is a tendency to apply the same condition to all impulse turbines, which has some justification, though in many cases axial thrust is not important so long as it is not excessive. A simple analytical discussion of the effect of blade-friction can be given when the entrance and exit angles are equal, which is useful of itself and as leading to the allowance for friction when the angles are unequal.

To allow for friction of steam in nozzles it is customary to assign a fraction  $\gamma$  to represent that portion of the heat which, through the effect of friction, is not changed into kinetic energy. Then the available heat for adiabatic action is multiplied by the factor  $1 - \gamma$  and the result is taken as the heat actually changed into kinetic energy.

To apply a complementary method to determining the effect of blade-friction we may assume that the exit velocity may be found by multiplying the entrance velocity by a factor

$$\sqrt{1 - \gamma}, \dots \dots \dots (1)$$

where  $\gamma$  now represents that portion of the kinetic energy at entrance which is changed back into heat by friction.

In Fig. 22  $ab$  represents the discharge velocity of the steam from a nozzle which makes the angle  $\alpha$  with the direction of motion of the wheel and its blades.

To find the relative velocity  $V_2$  of the steam entering the blades, lay off  $bc = V$ , the peripheral velocity of the wheel, and draw  $ac$ , then  $V_2 = ac$ . This construction gives also the entrance angle

$\beta$  for the blades, to give tangential entrance for the steam. If the entrance and exit angles are equal, that is if  $\gamma = \beta$ , then the exit velocity  $V_3$  may be laid off along  $ac$ , at  $ad$ .

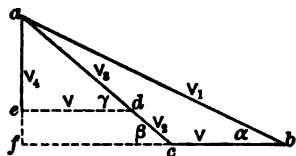


FIG. 22.

This relative velocity at exit may be computed by the equation

$$V_3 = \sqrt{1-y} V_2 \dots \dots \dots (2)$$

and laid off at *ad*. The absolute velocity  $V_4$  may now be determined by drawing the triangle *ade* with *de* equal to  $V$ , the peripheral velocity of the wheel. In order to have the best efficiency there should be no exit velocity of whirl; that is *ae* should be vertical, as shown in Fig. 22. The proper closure of the diagram with *e* on the vertical through *a* can be obtained graphically by trial, but the best way is to make a computation by the following method.

It is evident from Fig. 22 that

$$fb = V_1 \cos \alpha = fc + cb = fc + V. \dots \dots (3)$$

Again the triangles *ade* and *afc* give

$$cf : de :: ac : ad, \text{ or } cf : V :: V_2 : V_3; \text{ and } V_3 = V_2 \sqrt{1-y} \\ \therefore cf = \frac{V}{\sqrt{1-y}} \dots \dots \dots (4)$$

Substituting from equation (4) into equation (3) and solving for  $V$

$$V_1 \cos \alpha = V \left( 1 + \frac{1}{\sqrt{1-y}} \right) \\ \therefore V = \frac{\cos \alpha}{1 + \frac{1}{\sqrt{1-y}}} V_1 \dots \dots \dots (5)$$

This gives a direct solution for the velocity of the wheel when the velocity of the steam is known; conversely, if a velocity be assigned to the wheel the proper velocity at exit from the nozzle can be determined, and consequently the heat which should be assigned to the stage of the turbine in question.

To determine the angle  $\beta$  we have from Fig. 22,

$$\tan \beta = \frac{af}{fc} = \frac{af}{V \frac{1}{\sqrt{1-y}}} \text{ and } \tan \alpha = \frac{af}{bc + cf} = \frac{af}{V \left( 1 + \frac{1}{\sqrt{1-y}} \right)},$$

consequently  $\tan \beta = (1 + \sqrt{1 - \gamma}) \tan \alpha$  . . . . . (6)

It is proved on page 41 that the efficiency for an impulse turbine without friction and without velocity of whirl at exit is

$$e = \cos^2 \alpha. \quad . . . . . (7)$$

This efficiency can be analyzed as follows: The initial velocity of whirl is

$$V_w = V_1 \cos \alpha \quad . . . . . (8)$$

and as there is no resultant velocity of whirl the change of velocity or retardation is equal to the initial velocity of whirl, just written down. The best velocity for the wheel with equal entrance and exit angles is one-half the velocity of whirl, or

$$V = \frac{1}{2} V_1 \cos \alpha, \quad . . . . . (9)$$

and this is the distance through which the retarding force (or impulse) works. The work of one pound of steam is the product of the mass  $\left(\frac{1}{g}\right)$ , the retardation and the velocity, that is

$$\frac{1}{g} V_1 \cos \alpha \times \frac{1}{2} V_1 \cos \alpha = \frac{1}{2g} V_1^2 \cos^2 \alpha. \quad . . . . (10)$$

On the other hand, the kinetic energy per pound of steam in the jet is

$$\frac{1}{2g} V_1^2;$$

consequently the efficiency is

$$e = \frac{1}{2g} V_1^2 \cos^2 \alpha \div \frac{1}{2g} V_1^2 = \cos^2 \alpha.$$

Now the change in velocity of whirl for a blade with friction (as represented by Fig. 22) is unchanged and is represented by equation (8). But the impulse acts only through the distance

$$V = \frac{1}{1 + \frac{1}{\sqrt{1 - \gamma}}} V_1 \cos \alpha, \quad . . . . . (11)$$

as shown by equation (5), in one second. The work of one pound of steam with friction is

$$\frac{1}{g} V_1 \cos \alpha \times \frac{1}{1 + \frac{1}{\sqrt{1-y}}} V_1 \cos \alpha = \frac{1}{1 + \frac{1}{\sqrt{1-y}}} \frac{1}{g} V_1^2 \cos^2 \alpha. \quad (12)$$

The efficiency therefore becomes

$$e = \frac{2}{1 + \frac{1}{\sqrt{1-y}}} \cos^2 \alpha. \quad \dots \dots \dots (13)$$

The reduction in efficiency is chargeable entirely to slowing down the wheel to match the reduction of velocity of the steam in passing through the wheel. It is notable that the absolute velocity of the steam leaving the wheel  $V_4$  is less than it would be without friction; namely, it is represented by  $ae$  in Fig. 22 instead of  $af$ , as it would be were there no friction. This exit velocity has been restricted by our method to velocity of flow.

The discussion of blade-friction with equal entrance and exit angles on page 42 shows that such a choice of angles gives some axial thrust. We are now in condition to estimate the effect quantitatively. On Fig. 22 the axial retardation is represented by  $ef$ , while the peripheral retardation is represented by

$$fb = V_w = V_1 \cos \alpha.$$

From similarity of triangles  $aed$  and  $afc$

$$\begin{aligned} V_2 : V_2 - V_3 &:: V_4 : ef \\ \therefore ef &= (1 - \sqrt{1-y}) V_4 = (1 - \sqrt{1-y}) V_1 \sin \alpha. \end{aligned}$$

The ratio of the axial thrust to the impulse is

$$(1 - \sqrt{1-y}) V_1 \sin \alpha : V_1 \cos \alpha = (1 - \sqrt{1-y}) \tan \alpha. \quad \dots (14)$$

For example, if  $y$  be taken as 0.1 and  $\alpha$  as  $20^\circ$  then the ratio of the axial thrust to the impulse will be

$$(1 - \sqrt{1-0.1}) \tan 20^\circ = (1 - 0.95) 0.364 = 0.018.$$

Thus far the discussion of the effect of friction has been limited to blades with equal entrance and exit angles, but as shown on pages 40 and 42, there is an appreciable loss of efficiency from such a restriction if the kinetic energy of the velocity of flow is assumed to be lost; a discussion will be given later of the conditions in a pressure-compound turbine which is arranged so that the velocity of flow is not lost.

Some designers prefer to take advantage of the better efficiency that can be obtained by making the exit angle of the blades less than the entrance angle. To investigate this more general case refer to Fig. 23, in which the angle of the nozzle

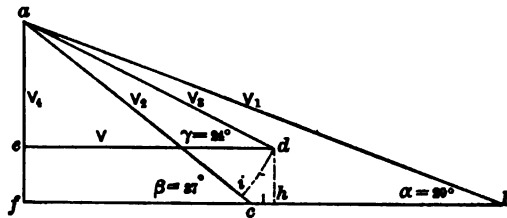


FIG. 23.

is  $20^\circ$  and the exit angle becomes  $24^\circ$ . The diagram is constructed by laying off  $V$  somewhat larger than half the velocity of whirl; in this case

$$V = 0.51 V_1 \cos \alpha.$$

Assuming  $\gamma = 0.1$ ,  $ai$  is laid off equal to

$$V_2 = \sqrt{1 - \gamma} V_1 = 0.949 V_1,$$

and then with a radius  $ai$  an arc is struck from  $a$ , intersecting the line  $hd$  at  $d$ ; the distance  $fh$  is also made equal to  $0.51 V_1 \cos \alpha$ , so that  $ed$  is equal to  $V$ .

As in the previous construction the diagram gives no velocity of whirl at exit, so that the retardation is equal to the initial velocity of whirl and the impulse per pound of steam is

$$\frac{1}{g} V_1 \cos \alpha.$$

The distance through which this impulse acts is  $V$ , so that the work of one pound of steam is now

$$\frac{0.51}{g} V_1^2 \cos^2 \alpha,$$

and the efficiency is

$$e = \frac{0.51}{g} V_1^2 \cos^2 \alpha \div \frac{1}{2g} V_1^2 = 1.02 \cos^2 \alpha = 0.901.$$

The efficiency for the same nozzle-angle and coefficient of friction, but with equal entrance and exit angle for the blades, will be

$$e = \frac{2}{1 + \frac{1}{\sqrt{1 - 0.1}}} \cos^2 \alpha = \frac{2 \times 0.9397^2}{1 + \frac{1}{\sqrt{0.9}}} = 0.860.$$

The gain from the smaller exit angle is therefore nearly five per cent.

The angles  $\beta$  and  $\gamma$  may be measured on the diagram or may be computed as follows:

$$\tan \beta = \frac{V_1 \sin \alpha}{V_1 \cos \alpha (1 - .51)} = \frac{0.3420}{0.46} = 0.743; \beta = 37^\circ,$$

$$V_s = \sqrt{0.9} V_2 = \{0.9 (0.3420^2 + 0.46^2)\}^{\frac{1}{2}} V = 0.559 V,$$

$$\cos \gamma = 0.51 \div 0.559 = 0.912; \gamma = 24^\circ 10'.$$

**Design of a Pressure Compound Turbine.** — Let the following conditions and data be assumed for the design of a pressure compound turbine:

Number of stages . . . . .	24
Brake horse-power . . . . .	8500
Gauge-pressure at steam-chest, pounds . . . . .	150
Vacuum, inches of mercury . . . . .	28
Revolutions per minute . . . . .	1000
Angles for nozzles, degrees . . . . .	20
Blade angles at entrance and exit equal	
Overall heat factor . . . . .	0.73
Friction factor for nozzles $\gamma_n$ . . . . .	0.05
Friction factor for blades $\gamma_b$ . . . . .	0.10
Factor for heat and rotation losses . . . . .	0.85
Mechanical efficiency . . . . .	0.94
Radiation and other losses . . . . .	0.03

The conditions assigned give the following absolute pressures, temperatures, and thermal properties if the atmospheric pressure is 14.7 pounds:

Pressures absolute.	Temperatures.	Entropy.	Heat contents.	Heat of liquid.
164.7 1	366 102	1.56 1.56	1193.3 871.1 <hr/> 322.2	.... 70

The thermal efficiency for Rankine's cycle is

$$e = 1 - \frac{C_2 - q_2}{C_1 - q_2} = 1 - \frac{871.1 - 70}{1193.3 - 70} = 0.287.$$

The steam per horse-power per hour for Rankine's cycle is, by equation (7), page 36,

$$W_R = \frac{2545}{C_1 - C_2} = \frac{2545}{322.2} = 7.89 \text{ pounds.}$$

The overall heat factor is 0.73, so that the steam per turbine horse-power per hour is

$$7.89 \div 0.73 = 10.8 \text{ pounds.}$$

The combined factor for mechanical friction and radiation is

$$0.94 \times (1 - 0.3) = 0.912,$$

consequently the steam per brake horse-power per hour is

$$10.8 \div 0.912 = 11.85 \text{ pounds.}$$

From the table on page 68 it can be inferred that the ratio for pressure distribution will be 1.055 for an overall heat factor 0.73. With this ratio the diagram for distribution factors can be drawn in Fig. 24. In this figure  $h_1 h_2$  represents the available heat 322.2 B.T.U. and can be divided into 24 parts for the 24 stages as shown.

This diagram may be used for two purposes: to find the ratio of the heat factor per stage to the overall heat factor, and to

find the distribution factors. The dotted line lettered 1.053, which is drawn at the mid-point of the first stage, is the distribution factor for that stage, and also the ratio of the factor per stage to the overall heat factor.

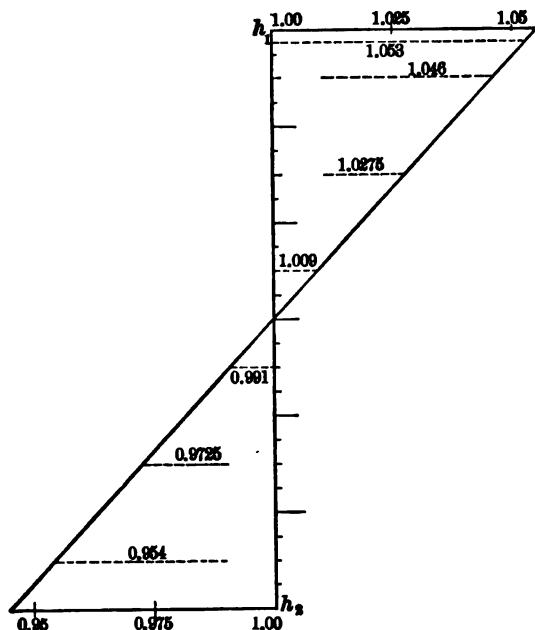


FIG. 24.

The heat factor per stage is the product of three factors, which take account of (1) the friction in the nozzle, (2) the blade efficiency as affected by friction, and (3) the disk friction and other similar losses. The first and last of these factors are indicated in the data of the problem. The second will now be determined by aid of equation (13), page 79, which gives in this case

$$e = \frac{2}{1 + \frac{1}{\sqrt{1 - y}}} \cos^2 \alpha = \frac{2 \times 0.9397^2}{1 + \frac{1}{\sqrt{0.9}}} = 0.860.$$

The heat factor per stage is therefore made up of the three

factors for the nozzle, the blade efficiency, and the heat and rotation losses,

$$(1 - 0.05) (0.860) (1 - 0.15) = 0.695.$$

The overall heat factor is

$$0.695 \times 1.053 = 0.73,$$

which shows that the several factors are concordant.

The adiabatic available heat for the entire range from  $366^{\circ}$  to  $102^{\circ}$  is 322.3 B.T.U., which would give for the first stage

$$322.2 \div 24 = 13.425 \text{ B.T.U.}$$

Multiplying by the distribution factor for this stage gives

$$13.425 \times 1.0527 = 14.13 \text{ B.T.U.}$$

The value of the friction factor for the nozzles is taken as 0.05, consequently the velocity of discharge is

$$V_1 = 223.7 \sqrt{14.13 \times 0.95} = 820.$$

This is the velocity of discharge from the nozzles of all the 24 stages.

The peripheral velocity of the pitch surface of the turbine, with equal angles for the blades and, without resultant velocity of whirl, will be in this case

$$V = \frac{\cos \alpha}{1 + \frac{1}{\sqrt{1-y}}} V_1 = \frac{0.9397}{1 + \frac{1}{\sqrt{0.9}}} 820 = 375.$$

Since there are to be 1000 revolutions per minute the diameter of the pitch surface will be

$$d = \frac{375 \times 60}{1000\pi} = 7.16 \text{ feet} = 7 \text{ feet } 2 \text{ inches nearly.}$$

The blade angles may be computed by equation (6), page 78:

$$\begin{aligned} \tan \beta &= (1 + \sqrt{1-y}) \tan \alpha = (1 + \sqrt{0.9}) 0.3640 = 0.710, \\ \beta &= 35^{\circ} 20'. \end{aligned}$$

To proceed with the determination of dimensions of nozzles and blades it is necessary to find the specific volumes of the steam at the nozzles by the usual process of distributing the intermediate temperatures and pressures. If this process is applied to the 24 stages marked off on  $h_1h_2$ , Fig. 24, it will be very tedious, especially as it will be necessary in that case to interpolate in the entropy table to tenths of a degree. The process may be abbreviated by determining the temperature and quality for each fourth stage, after which the intermediate quantities can be readily determined. For this purpose let the line  $h_1h_2$ , in Fig. 24, be divided into six parts as shown, and let abscissæ be drawn at the mid-points, on which the distribution factors can be read. In practice the work can be done by aid of a sketch, since the factors are readily computed. Thus, the factor for the mid-point of the first four stages is

$$1 + \frac{1}{6} \times 0.055 = 1.046.$$

The adiabatic available heat for four stage intervals is

$$322.2 \div 6 = 53.7,$$

and this quantity multiplied by the factors from Fig. 24 gives

$$\begin{aligned} 53.7 \times 1.046 &= 56.2 \\ 53.7 \times 1.0275 &= 55.2 \\ 53.7 \times 1.009 &= 54.2 \\ 53.7 \times 0.991 &= 53.2 \\ 53.7 \times 0.9725 &= 52.2 \\ 53.7 \times 0.954 &= 51.2 \end{aligned}$$

322.2 total.

It is important that the portions should be distributed so as to give the sum equal to the total available heat and that cumulative errors should not be allowed to enter the work, since they would be applied to the last stage and derange the computation for that stage.

In the following table the heat portions just determined are

subtracted in succession to determine the heat contents at each stage; the temperatures and pressures corresponding can be found in the temperature-entropy table at entropy 1.56. The heat changed into work per stage is

$$53.7 \times 0.73 = 39.2,$$

where 0.73 is the overall heat factor. This quantity is subtracted six times in succession to find the heat contents of the steam approaching the nozzles for the stages entered in the table. The corresponding qualities are interpolated in the entropy table at the proper temperatures. If we were determining volumes for all twenty-four stages we should find volumes directly, but for the abbreviated computation it is better to find the quality, because that property changes slowly.

TEMPERATURES, PRESSURES, AND QUALITY.

Stage.	Heat contents, entropy 1.56.	Temperatures.	Pressures.	Heat contents, factor 0.73	Quality.
0	1193.3 56.2	366	164.8	1193.3 39.2	0.999
4	1137.1 55.2	313.5	81.7	1154.1 39.2	0.969
8	1081.9 54.2	265	38.5	1114.9 39.2	0.943
12	1027.7 53.2	220	17.2	1075.7 39.2	0.920
16	974.5 52.2	178	7.19	1036.5 39.2	0.899
20	922.3 51.2	138.5	2.78	997.3 39.2	0.878
24	871.1	102.0	1.00	958.1	0.858

Having the temperature and pressure at each fourth stage and in the exhaust ( $102^{\circ}$ ) the diagram shown by Fig. 25 may be drawn with the number of stages as abscissæ and with the corresponding qualities from the preceding table as ordinates; from this curve the qualities for all the remaining stages may be read and entered in the table on page 88. A similar curve could be drawn for the temperatures, but in order to get the precision of one or two tenths of a degree, as is desirable for finding volumes, a large scale would be required. To obtain the

desired precision with a compact diagram the following device can be used. It may be noticed that in the preceding table

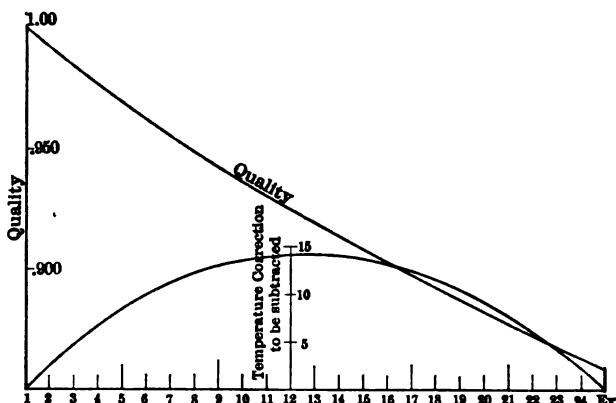


FIG. 25.

the temperatures are spaced unevenly; a uniform distribution would give the following:

Stages.	0	4	8	12	16	20	24
Uniform distribution . . .	366	322	278	234	190	146	102
Actual distribution . . . .	366	313.5	265	220	178	138.5	102
Difference . . . . .	0	8.5	13	14	12	7.5	0

The difference between the actual temperatures and uniformly distributed temperatures are plotted in Fig. 25, and from the curve corrections may be read for uniformly distributed temperatures for all the stages. In this way the temperatures are filled in for the table on page 88.

The specific volumes in that table are found by multiplying the specific volume of saturated steam by  $x$  instead of using the more precise equation

$$v = xu + \sigma,$$

because either  $x$  is nearly unity or  $u$  is large compared with  $\sigma$ .

## TEMPERATURES, PRESSURES AND VOLUMES.

	Tem- pera- ture, <i>t</i>	Pressure absolute, <i>p</i>	Qual- ity, <i>z</i>	Volume, <i>z</i>		Tem- pera- ture, <i>t</i>	Pressure absolute, <i>p</i>	Qual- ity, <i>z</i>	Volume, <i>z</i>
	366.0	164.8	0.999	....	13	209.2	13.90	0.914	25.8
1	352.6	140.1	0.992	3.20	14	198.5	11.18	0.909	31.4
2	339.4	117.0	0.984	3.75	15	188.0	8.95	0.904	38.5
3	326.4	98.0	0.977	4.36	16	178.0	7.19	0.899	47.0
4	313.5	81.7	0.965	5.17	17	168.0	5.72	0.893	57.7
5	301.0	68.0	0.962	6.12	18	158.0	4.52	0.888	71.7
6	288.8	56.5	0.956	7.25	19	148.1	3.54	0.883	89.6
7	276.8	46.7	0.949	8.60	20	138.5	2.78	0.878	111.8
8	265.0	38.5	0.943	10.25	21	129.2	2.17	0.873	140.0
9	253.4	31.6	0.937	12.25	22	120.0	1.69	0.868	176.0
10	242.1	25.9	0.931	14.65	23	111.0	1.31	0.863	222.5
11	231.0	21.2	0.925	17.60	24	102.0	1.00	0.858	284.5
12	220.0	17.19	0.920	21.3	...	....	....	....	....

The steam per brake horse-power per hour has been computed to be 11.85 pounds; as there are to be 8500 brake horse-power this gives

$$11.85 \times 8500 \div 60^2 = 28$$

pounds per second; this is the weight to be used in calculating the areas of the nozzles and the passages between the blades.

The velocity  $V_1$  for all the nozzles has been computed to be 820 feet per second and the specific volume of the steam discharged through the last nozzles into the last chamber which is connected directly with the exhaust is 284.5 cubic feet, as set down in the preceding table. Consequently the net area in square inches is

$$284.5 \times 28 \times 144 \div 820 = 1400.$$

In order to arrive at the dimensions of the nozzles for a turbine after the net area has been computed, it is necessary to take account of (1) the angle which the jet makes with the face of the wheel, (2) the thickness of the plate or guide separating the nozzles, and (3) of the blank spaces (if any) which are required for assembling the diaphragms or the turbine casing. Turning to Fig. 26, page 89, it is clear that the gross width of a nozzle  $ac$  will be equal to the pitch of the nozzles  $ab$  multiplied by the sine of the angle. From the gross width  $ac$  there must

be subtracted a fraction to allow for the thickness of the guide at  $a$ ; this fraction may vary from one-thirtieth to one-tenth, depending on the angle and the construction of the nozzles. In Fig. 26 it is one-twentieth. The allowance for blank spaces for assembling will be determined from the assembly drawings of the turbines. In the present case the blank spaces will be assumed to take up 3.8 inches.

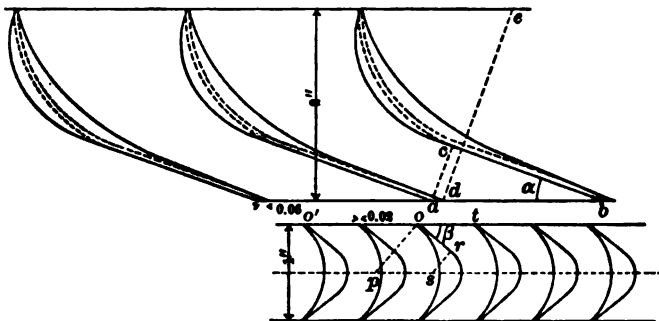


FIG. 26.

The pitch diameter of the turbine has been determined to be 7 feet and 2 inches; the perimeter is

$$86 \times 3.1416 = 270.2 \text{ inches;}$$

subtracting 3.8 inches and also one-thirtieth for the thickness of the guides gives for the total available width of all the orifices,

$$(270.2 - 3.8) \frac{1}{30} \sin 20^\circ = 255 \times 0.342 = 87.2 \text{ inches.}$$

Consequently the radial dimension or length of the nozzles for the last stage will be

$$1400 \div 87.2 = 16.05 \text{ inches.}$$

For various reasons this length is inconvenient, and in practice one of several devices would be employed to reduce it, but for simplicity we will for the present accept the length and base the computation of other lengths on it. The length of the clear space in the blades between the root and the crown will be made equal to the radial length of the nozzles; consequently the

lengths computed may be called the *blade lengths*. These will be the blade lengths at entrance; in order to allow for the effect of blade friction the lengths at exit will be slightly increased.

Since the steam velocity of the jet for all the stages is assumed to be the same, the blade lengths will be inversely proportional to the specific volume as set down in the table on page 89; thus the blade length for the twenty-third stage is

$$16.05 \times \frac{222.5}{284.5} = 12.55,$$

and other lengths in the following table are found in the same way.

BLADES AND NOZZLES.

Stages.	Number of nozzles.	Blade lengths.		Stages.	Number of nozzles.	Blade lengths.	
		Entrance.	Exit.			Entrance.	Exit.
1	38	0.703	0.741	13	148	1.46	1.54
2	44	0.712	0.750	14	148	1.77	1.87
3	52	0.700	0.738	15	148	2.17	2.30
4	62	0.698	0.736	16	148	2.65	2.81
5	74	0.691	0.728	17	148	3.26	3.45
6	86	0.704	0.742	18	148	4.05	4.28
7	100	0.718	0.757	19	148	5.06	5.34
8	120	0.713	0.752	20	148	6.31	6.68
9	148	0.691	0.732	21	148	7.90	8.36
10	148	0.826	0.872	22	148	9.94	10.45
11	148	0.993	1.05	23	148	12.55	13.40
12	148	1.20	1.27	24	148	16.05	17.05

For a turbine of this type it is customary to assign a minimum blade length; when this has been reached the admission becomes partial, the number of nozzles being selected to give the proper total area. The minimum length for this design may be taken as about 0.7 of an inch. Since fractional numbers cannot be assigned to nozzles, the radial dimension can be adjusted to give the proper total area. Partial admission here begins with the eighth stage. The construction to be given later for the forms of blades and nozzles indicates that 148 nozzles may be used for full peripheral admission. If, then, there are 120

nozzles assigned to the eighth stage the blade length may be computed as follows:

$$\frac{16.05 \times 10.25 \times 148}{284.5 \times 120} = 0.713 \text{ inch,}$$

where 284.5 is the specific volume for the discharge of the twenty-fourth set of nozzles, and 10.25 for the discharge of the eighth set of nozzles, while the longest blades are 16.05 inches long; the reduction from 148 to 120 nozzles increases the length in the inverse ratio of those numbers. The number 120 was selected as that even number which would come nearest to giving a blade length of 0.7 of an inch. In like manner the number of nozzles was selected for all the remaining stages and the blade lengths at entrance were computed.

The blade friction changes some of the kinetic energy into heat which dries and expands the steam, and the exit velocity is consequently less than the entrance velocity. Under some circumstances the drying of the steam has an appreciable effect, but here it can be shown to be of no importance.

From Fig. 22, page 76, it is evident that the entrance velocity to the blades is

$$V_2 = V_1 \frac{\sin \alpha}{\sin \beta} = 820 \frac{\sin 20^\circ}{\sin 35^\circ 20'} = 485 \text{ feet.}$$

The velocity due to a certain available heat is computed by the equation

$$V = 223.7 \sqrt{\text{heat}};$$

conversely the heat corresponding to a given velocity can be computed by the equation

$$\text{heat} = \left( \frac{V}{223.7} \right)^2,$$

which gives in this case

$$\left( \frac{445}{223.7} \right)^2 = 4.7 \text{ B.T.U.}$$

One-tenth of this is the heat resulting from friction; namely,

about half a thermal unit, which can have only an insignificant effect.

The velocity at exit from the blades is

$$V_3 = \sqrt{1 - \gamma} V_2 = \sqrt{0.9} V_2 = 0.949 V_2;$$

since the exit areas from the blades should be inversely proportional to the velocity, the blade lengths at exit, given in the table on page 89, can be found by dividing the entrance lengths by 0.949.

**Construction of Blades and Nozzles.** — The forms to be given to the blades and nozzles depend in part on the conditions of the design, and in part on the methods of construction, but much is left to the discretion of the designer. The common methods of drawing the contours will be explained, and certain general proportions will be given for the guidance of the student; an experienced designer will not hesitate to depart from the conditions stated here at discretion.

Fig. 26 shows the construction for the blades and nozzles of the design just computed. The axial width of the blades is taken as one inch and the pitch is determined to be 0.6 of an inch. The line  $or$  is drawn making the angle  $\beta$  equal to  $35^\circ 20'$  as stated on page 84; the line  $op$  is drawn perpendicular to  $or$  thus locating the center  $p$  from which the face of the blade is drawn. The pitch of the blade is laid off from  $p$  to  $s$ , and  $s$  is the center from which the back of the blade is drawn. The edge of the blade is made 0.02 of an inch wide measured along the line  $oo'$ . This construction gives a passage of uniform width through the blades.

The pitch of the nozzles is made 1.8 inch, that is there are three blades for each nozzle. The axial dimension of the nozzle is taken as two inches; that having been found by trial to give a good form to the passages through the nozzles.

The line  $bc$  is laid off making the angle equal to  $20^\circ$ , as required by the conditions of the design, and  $de$  is drawn perpendicular to  $bc$ , to locate the center  $c$  from which an arc is drawn for constructing the upper end of the plate or guide; this gives an axial

entrance for the steam to the nozzle which is proper when there is no exit velocity of whirl from the preceding wheel. The width  $ad$  of the plate measured along  $ab$  is made equal to 0.06 of an inch, which corresponds to the width of the edges of three blades.

Two constructions are shown for the nozzles; one in dotted lines is a proper construction when the nozzles are made of plates of uniform thickness; the other construction shows the division plates thickened in the middle. The second construction is made with smooth curves, drawn so as to give a contracting passage which does not change the direction of flow at exit.

The perimeter of the pitch surface has been found to be 270.2 inches; with 450 blades this gives the pitch

$$270.2 \div 450 = 0.60044$$

of an inch. There being one nozzle for three blades, we should have a possible number of 150 nozzles if no allowance were required for assembling the diaphragm; it has already been assumed that 148 nozzles may be used for full admission, thus assigning 3.8 inches for assembling.

In laying out a new design it is convenient to begin by selecting an axial width of the blade which must be sufficient to ensure strength and stiffness; a computation for strength will be given later, but it is probable that experience is the best guide in this matter. As a rough guide the width may be made about one-tenth to one-twelfth of the greatest blade length. For the design just given this would call for a width of 1.6 of an inch, but as already said, some method would probably be found in practice to reduce that length, and we have selected one inch as a convenient width of blade for the design.

Having the width of the blade a pitch must be selected which will give a good construction for the blade and also a convenient number of blades for construction of the wheel. To aid the student in selecting a pitch several forms of blades have been drawn as shown in Fig. 27. The nozzle angle has been assumed

to vary from  $15^\circ$  to  $30^\circ$  and for each angle an appropriate ratio of pitch to width of blade has been selected; the selection has purposely been made somewhat crudely so that attention may be called to the fact that the forms are a rough guide only. In criticism of the forms in Fig. 27 it may be said that the form for  $20^\circ$  nozzles is probably good; the form for  $15^\circ$  gives a very narrow passage; the form for  $25^\circ$  may in practice be as good as that for  $20^\circ$ , but the form for  $30^\circ$  gives a thin blade which is unduly humped up on the back. An attempt may be made to improve the form for  $15^\circ$  by increasing the pitch, but at the expense of humping up the back; on the contrary we may reduce the hump for  $30^\circ$  by decreasing the pitch, but the blade will then become thinner than it now is. Small changes of pitch will, of course, have slight effect on the form of the blade.

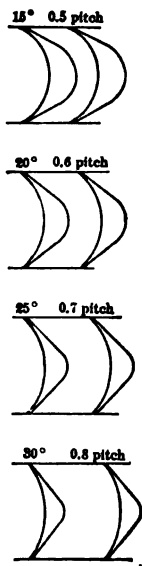


FIG. 27.

Some thickness must be given to the blades at the edge, depending on the size of the blade and the metal of which it is made; the peripheral width may be from 0.02 to 0.04, the least being proper for moderate widths and strong, hard material like steel or bronze. It has been found undesirable to give a sharp edge to a bucket even when made of steel.

The Rateau pressure compound turbine is sometimes given blades that are stamped from sheet steel; in such case the pitch will be made smaller than has been recommended and the faces only will be formed to guide the steam. A compromise between this method and that shown by Fig. 27 may be had by humping up the backs less and accepting the resulting change in the width of passage. For a hydraulic turbine such an increase and decrease of effective area of a passage, if not too great, has little or no effect on efficiency, but since steam is an elastic fluid there is liable to be a loss of efficiency, because it involves a change from a given temperature to a lower temperature and back

again, which changes are not thermodynamically reversible. It is difficult to estimate the loss on this account for steam and it may not be important for a turbine of the type under consideration.

It will in general be found convenient and sufficient to have about one-third as many nozzles as there are blades, the number depending partly on the nozzle angle and partly on the construction. If fewer nozzles are chosen a greater axial width will be required for good form; the ratio width to pitch may be from 1.5 to 3.

If the radial length of the nozzles and the entrance blade length are made equal, there should be a certain relation between the peripheral dimension  $ab$  of a nozzle, Fig. 26, and the corresponding width of the edge of the blade. In the figure mentioned there are three blades for each nozzle and the nozzle-dimension is three times the width of blade edge. To understand this relation consider that the velocity of flow out of the nozzle and into the blades is the same; it is represented by  $af$ , Fig. 22, page 79. Consequently the area of the section of a nozzle by a plane parallel to the face of the wheel must be equal to the sum of the areas between the corresponding blades by a similar plane. Thus, if the radial lengths of a nozzle and the blades are equal, the line  $bd$ , Fig. 26, should be three times the length of the line  $ot$ ; there being three blades for a nozzle. A different ratio for the number of blades and the number of nozzles would give a different factor; thus if there were 3.5 blades per nozzle that factor would be used in place of 3. The same relation must exist for the blank spaces due to the thickness of the wall of the nozzle and the edges of the blades. Should it be found convenient to change the ratio of the blank spaces mentioned, then allowance must be made in the relation of the radial dimensions of the nozzles and blades. For example, if the edges of the blades are thickened, then the length must be increased; but if the wall of the nozzle is thickened it will not be wise to decrease the blade height at entrance. In such case the blade height may be computed, making the proper allowance, and therefrom the height at exit may be found, but the blade

height at entrance will be made equal to the length of nozzle at exit, even though that is more than the computed length.

Suppose that the blade thickness at the edge measured on  $oo'$ , Fig. 26, is  $\frac{1}{m}$  part of the pitch of the blades, and that the dimension  $ad$  is  $\frac{1}{n}$  part of the pitch of the nozzles; then the blade height at entrance should be

$$\left(1 - \frac{1}{n}\right) \div \left(1 - \frac{1}{m}\right)$$

times the radial length of the nozzle. Thus if the edge of the blades were 0.03 instead of the dimension given, the blades would have the ratio

$$0.03 \div 0.60 = \frac{1}{20},$$

while the nozzles would have the ratio

$$0.06 \div 1.8 = \frac{1}{30};$$

the blade height would therefore be

$$\left(1 - \frac{1}{30}\right) \div \left(1 - \frac{1}{20}\right) = 1.017$$

of the length of the nozzle. Applying this factor to the first stage of the table on page 90 would give

$$0.703 \times 1.017 = 0.713$$

for the blade height at entrance; the blade height at exit would be increased in like proportion.

**Reducing Blade Lengths.** — As appeared in the design just discussed the blade lengths for the lowest stages of a pressure compound turbine may become inconvenient. Of course the length may be accepted and the width required to give sufficient strength may be adopted for those low pressure stages, while narrower blades may be used with the higher stages. In such case the number of blades will be correspondingly reduced.

The most ready way of reducing the length is to assign a larger nozzle angle to the lower stages. The length will be nearly

inversely as the sine of the angle, so that a nozzle angle of  $30^\circ$  instead of  $20^\circ$  will reduce the length in the ratio of

$$\sin 20^\circ \div \sin 30^\circ = 0.3420 \div 0.5000 = 0.684;$$

applied to a blade length of 16.05 inches this will reduce it to about 11 inches.

Now the peripheral velocity should be the same for all stages of a turbine and consequently a larger nozzle angle will call for a greater velocity of discharge. If  $V_1$  is the discharge velocity for the higher stages, then  $V_1'$  for the lower stages may be computed from equation (5), page 77, as follows:

$$V_1' = \frac{1}{\cos \alpha'} \left( 1 + \frac{1}{\sqrt{1-y}} \right) V = \frac{\cos \alpha}{\cos \alpha'} \frac{1 + \frac{1}{\sqrt{1-y}}}{1 + \frac{1}{\sqrt{1-y}}} V_1.$$

$$\therefore \frac{V_1'}{V_1} = \frac{\cos \alpha}{\cos \alpha'} = \frac{\cos 20^\circ}{\cos 30^\circ} = \frac{0.9397}{0.8660} = 1.085;$$

the consequence of this increase of velocity will be to reduce the blade length mentioned above from 11 inches to 10.1 inches.

The approximate methods just given are sufficient to show that the blade lengths may readily be reduced by increasing the nozzle angle, but if it is decided to use larger nozzle angles for the lower stages, a complete new design must be made, assigning larger portions of heat to those stages. In general the blade friction factor  $y$  will be the same so that the tacit assumption in the preceding equation will be justified. The amount of heat for stages with different nozzle angles will be proportional to the squares of the velocities, so that with  $20^\circ$  and  $30^\circ$  the heat portion for a stage with the larger angle will be

$$\frac{\cos^2 \alpha}{\cos^2 \beta} = \left( \frac{0.9387}{0.8660} \right)^2 = 1.175.$$

If the increased nozzle angle is applied to the two last stages of a 24-stage turbine then the line  $h_1h_2$  of Fig. 24, page 83, must

be divided into unequal parts, the two lowest parts being 1.175 times as long as the others. Starting from this point the design may be carried out much as in the manner given.

Another way of avoiding excessive blade lengths will be to increase the pitch diameter. Some pressure compound turbines have two or three pitch diameters, the ratios being about,

$$1 : 1.4,$$

$$1 : 1.2 : 1.5.$$

The peripheral velocities will be in the same ratio, and the heat assignments will be as the squares, that is in the ratios

$$1 : 1.96$$

$$1 : 1.44 : 2.25,$$

provided that the nozzle angles are constant.

**Clearance.** — Since the intention is to expand the steam in a certain set of nozzles down to the pressure in the chamber to which they discharge, the radial clearances of the wheels of a pressure compound turbine can be made as large as may be desirable for construction.

It is desirable to give a considerable axial clearance of the blades in a wheel from the nozzles, as the blade edges will probably give less resistance than if the clearance is as small as construction will allow. The clearance may be made from one-tenth to one-fourth of the width of the blade.

**Shaft-Leakage.** — It is customary to make the diaphragms for a pressure compound turbine with a small clearance at the shaft, or more properly at the hub of the wheel; this clearance may be 0.001 to 0.002 of an inch. To estimate the leakage of the turbine discussed on page 81, consider that the effective area for all the nozzles of the second stage of the turbine discussed on page 81 is

$$3.75 \times 28 \times 144 \div 820 = 18.4$$

square inches; the factors are in order, the specific volume of the steam discharged from the second set of nozzles, the weight

of steam in pounds per second, and the number of square inches in a square foot; the divisor is the velocity of discharge from the nozzles in feet per second.

The hub of the wheel may have a diameter of about 19 inches, so that the perimeter is about 60 inches. If the clearance is 0.015 of an inch the area for leakage will be

$$60 \times 0.015 = 0.9$$

of an inch. If the flow of steam through the clearance were as free as through the nozzles, then the leakage would be about five per cent. But experiments on flow through straight, long tubes show that the discharge is likely to be half the theoretical free discharge, or even less. It is the habit to groove the surface of the diaphragm, thus forming a kind of a labyrinth packing, and the leakage may therefore be estimated as two per cent or less. Allowance for leakage may be made, if thought proper, by reducing the radial length of the nozzles of the first diaphragm by one or two per cent; the allowance for the succeeding diaphragms may be made progressively smaller as the specific volumes increase, and after four or five diaphragms have been considered the rest may be left as designed.

**Radiation.** — The external radiation and other losses are estimated as two or three per cent of the steam consumption. All this steam passes through the first set of nozzles, and that part not condensed in the first stage passes through the second set of nozzles, etc. It therefore appears proper to increase the radial length of the nozzles to allow for this action. Though there is no information concerning the rates of radiation from different parts of the turbine, it must be more energetic near the high-pressure end, and this is the end where the effect of shaft leakage is greatest. Perhaps we may offset one against the other and take the first computation of blade lengths as sufficient. This, of course, cannot apply to the first set of nozzles, but the power of the turbine is commonly regulated by adjusting the number or area of those nozzles, in which case a margin of three per cent will not be important.

**Methods of Fixing Blades.** — A number of methods of fixing blades to the wheels of a pressure compound turbine have been devised, of which a few will be shown here; other methods that are available will be found on page 168 in connection with the description of pressure and velocity compound turbines.

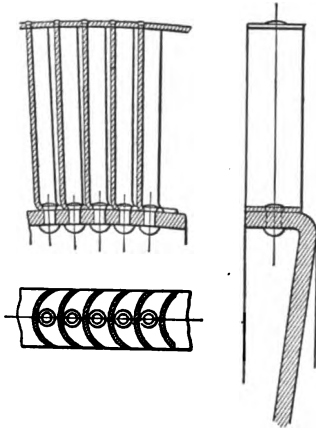


FIG. 28.

A method of making and fixing blades for the Rateau turbine is shown by Fig. 28. The wheels were pressed from steel plates, and the blades were stamped to form from sheet steel, and riveted to the flange of the wheel. The pitch of the blades is about half their width, so that they are relatively numerous, and the faces only are shaped to the correct form.

The blades for the Zoelly turbine are shaped much like those for the Rateau turbine, but they are worked from solid steel and are set in a groove as shown by Fig. 29. One side of the groove is a ring which is riveted to the edge of the wheel.

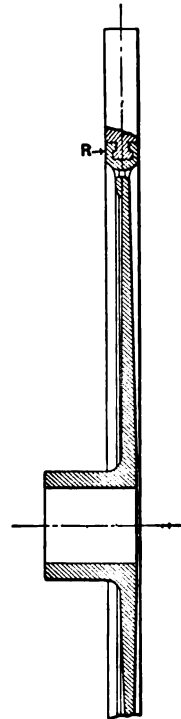


FIG. 29.

One of the blades for a turbine of the Rateau type, recently built by the Westinghouse Company for the Greenwich power station, is shown by Fig. 30. It is machined from nickel steel and slotted to straddle the edge of the wheel to which it is riveted. A tit at the end passes through a band or crown and is then riveted over.

**Construction of Nozzles.** — The converging nozzles proper for a pressure compound turbine with many stages are com-

paratively simple to make. They may be separated simply by metallic plates pressed to form as indicated by the dotted lines of Fig. 26, in which case the ends of the plate are fixed in or cast in the edge and body of the diaphragm. When this construction is used it is likely that some of the plates must be made extra heavy or that in some way rigid connection of the

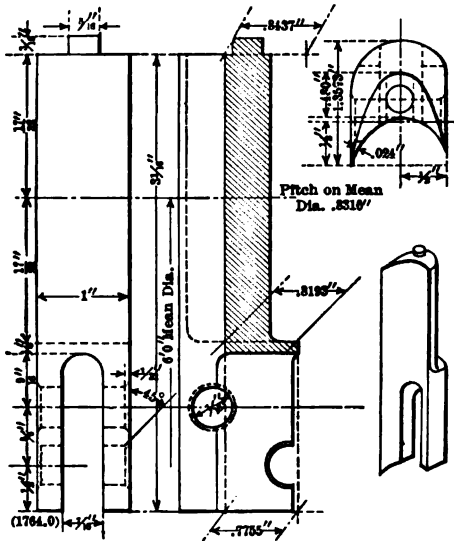


FIG. 30.

edge and body must be secured. If the separating pieces between the nozzles are thickened as shown by the full lines, they may give sufficient rigidity.

Fig. 31 shows the forms of the first nozzles of the Greenwich turbine. They are arranged in groups and worked out of one casting. Valves moved by a hand wheel are arranged for adjusting the area to the power required from the turbine. The construction is further illustrated by Fig. 32.

This turbine is designed to run at a relatively low peripheral velocity and the steam from the blades has a reversed velocity of whirl; in consequence the nozzles have their openings directed



of nozzles is increased from stage to stage. When the space is considerable it is probable that the velocity of discharge from the wheel is dissipated and the kinetic energy changed into heat, so that the steam enters the next nozzles with little or no initial velocity, as has been so far tacitly assumed in this chapter. If

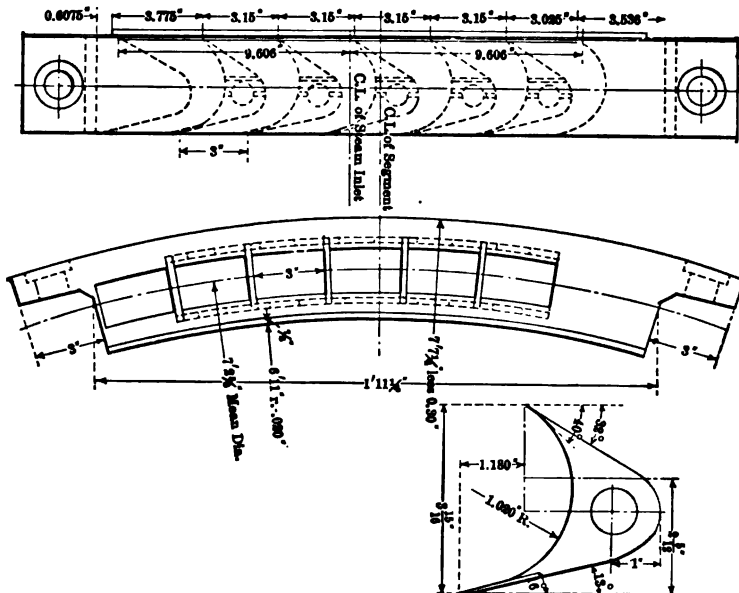


FIG. 33.

the velocity of discharge is dissipated it is a matter of little consequence where the nozzles are placed. But if the nozzles are near enough to consume a considerable part of the energy due to the velocity of discharge, the nozzles should be so placed that the steam will enter them directly.

Let  $abcde$ , Fig. 34, represent a blade which has steam entering tangentially with the velocity  $V_2$ , while the blade itself has the velocity  $V$ . Assuming that the relative velocity is constant, we may divide

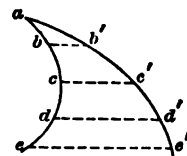


FIG. 34.

the curve into a number of equal parts that are approximately straight. From  $b$  lay off

$$bb' = ab \frac{V}{V_2},$$

then  $b'$  will be a point in the path of a particle of steam. In like manner

$$cc' = 2 ab \frac{V}{V_2}, \text{ etc.}$$

The curve  $ab'c'd'e'$  may be taken as the path of the steam through the wheel and  $ee'$  is the *lead*, that is, the distance that the nozzle should be set forward, provided that the blades are designed to give no velocity of whirl, because the velocity of discharge is thus directed vertically downward in the figure. If there is a velocity of whirl at exit, then the direction of the absolute exit velocity  $V_4$  must be determined as in Fig. 10, page 45, and a line drawn in that direction from  $e'$  will indicate the proper location of the nozzle and also the angle of entrance to it.

**Strength of Blades.** — The computation for strength and stiffness of a blade will depend on the method of fixing the blade to the wheel. If the blades are well secured, as shown by Fig. 30, then the full strength of the blade may be developed; if a blading like that on page 168 is used then the root of the blade only can be considered in determining the resistance to centrifugal force and bending.

Though the blades are commonly fixed to a band or crown at the ends, it is better to treat them as cantilevers bending about an axis at the root. The section at the root is subjected to direct stress from centrifugal force and to bending from impulse; if the exit angle is much smaller than the entrance angle, it may be necessary to inquire into the axial thrust due to retardation of flow, but that influence is usually unimportant.

As an illustration of the methods, a computation will be made for the longest blade for the design on page 79. The

section of the blade shown on page 89 has the following properties:

Area, square inch . . . . .	0.6
Length, inches, mean . . . . .	16.5
Weight, cubic inch steel, pound . . . . .	0.28
Total, pounds . . . . .	2.8
Moment of inertia. . . . .	0.0058
Distance most strained fiber, inch . . . . .	0.32

The centrifugal force may be computed with sufficient approximation by multiplying the mass by the square of the peripheral speed ( $V = 375$ ) and dividing by the radius (3.58 feet). This gives

$$\frac{2.8 \times 375^2}{32.2 \times 3.58} = 3410 \text{ pounds.}$$

There is, therefore, a direct stress of

$$3410 \div 0.6 = 5690.$$

The turbine is to develop 8500 brake horse-power with a mechanical efficiency of 0.94, so that the gross horse-power or turbine horse-power is

$$8500 \div 0.94 = 9040.$$

The power is equally divided among 24 stages so that there is

$$9040 \div 24 = 377$$

horse-power per stage. There are 450 blades on the low pressure wheel but six are inoperative on account of blank spaces in the diaphragm where it is assembled. Consequently the horse-power per blade is

$$377 \div 444 = 0.85.$$

Now the peripheral velocity is 375 feet per second and if the impulse is  $F$  pounds the corresponding horse-power is

$$\frac{375 F}{550} = 0.85.$$

$$\therefore F = \frac{0.85 \times 550}{375} = 1.25 \text{ pounds.}$$

Since the blade is 16.05 inches long the bending moment in inch-pounds is

$$\frac{1}{2} \times 1.25 \times 16.5 = 10.3.$$

The equation for computing stress is

$$f = \frac{My}{I} = \frac{10.3 \times 0.32}{.0058} = 568,$$

where  $M$  is the bending moment,  $y$  is the distance of the most strained fiber from the neutral axis, and  $I$  is the moment of inertia. The total stress is therefore

$$5690 + 568 = 6260 \text{ pounds.}$$

The centrifugal force appears to have the principal influence in producing stress in the blades and the bending, though it has an appreciable effect, is of secondary importance. The centrifugal stress, as computed above, is directly proportional to the length of the blade and independent of the area. Consequently if the stress appears to be excessive in any case the length must be reduced. Reducing the length also reduces the stress due to bending; it might be reduced by increasing the width of the blade, but as that does not affect the centrifugal stress it cannot be considered an effective way of reducing stress.

The centrifugal force acting on the blades is proportional to the square of the peripheral velocity and inversely proportional to the radius of the pitch surface; it can therefore be reduced by reducing the peripheral speed or by increasing the diameter of the turbine pitch surface.

The provision for overload of a turbine is commonly made by introducing nozzles that deliver steam expanded from boiler pressure onto the blades of one of the lower stages. The impulse of the steam is intense from the fact that the velocity is high and the peripheral velocity relatively low. The blades for such a stage must be made either of steel or strong bronze and must be well secured. The impulse is to be computed from the weight of steam discharged and the retardation as explained in Chapter II.

**Unbalanced Impulse.** — If the admission of steam is restricted to a part of the periphery of a turbine, the nozzles may be arranged in two or more symmetrical groups, in which case the impulse from the nozzles will be balanced, or they may be put in a single group to reduce the spreading of the steam. If the latter arrangement is chosen an investigation of the unbalanced impulse may be desired, because it will be transferred to the bearings. In Fig. 35 let  $be$  and  $b'e'$  be the impulse acting on two blades equidistant from the line  $ox$ ; resolving into component parallel and perpendicular to  $ox$  we shall have the components  $bc$  and  $b'c'$  balanced so far as bearing-pressure is concerned, but the components  $bd$  and  $b'd'$  are unbalanced and their sum will be borne by the bearings of the shaft. These components are each equal to the impulse multiplied by the cosine of the angle  $box$ . To find the resultant pressure of the bearings due to any number of unsymmetrical nozzles in a group, we may multiply the total impulse by the mean cosine of the half angle subtended by the group. The mean cosine may be found by summing up the cosines for the individual nozzles and dividing by the number of nozzles. But as this may be a tedious process and as precision is not important, we may take instead of the mean cosine the cosine of a quarter of the angle  $bob'$ , Fig. 35. For an angle of  $bob' = 180^\circ$ , this will overestimate the effect ten per cent; if desired we may use this as a basis for computing the total increase of bearing-pressure for a turbine.

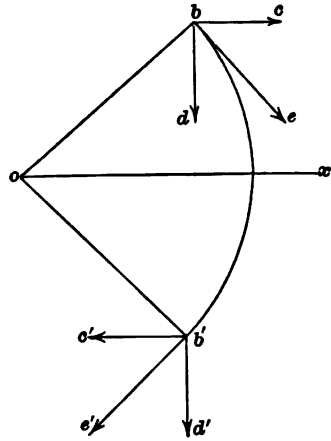


FIG. 35.

The details of a computation can be shown by applying it to the turbine discussed on page 81. The table on page 90 shows the number of nozzles for the earlier stages which are given in the following table:

No. of stage.	No. of nozzles.	Arc of admission.	Unbalanced arc.	Quarter angle.	Cosine.	Impulse.	Bearing pressure.
1	38	91	91	23	0.921	553	511
2	44	105	105	26	0.899	553	497
3	52	125	125	31	0.857	553	474
4	62	149	149	37	0.799	553	442
5	74	177	177	44	0.719	553	397
6	86	206	154	39	0.777	413	321
7	100	240	120	30	0.866	276	239
8	120	288	72	18	0.951	136	129
9	148	...	...	...	...		
						Total	3010

In addition to the 148 nozzles set down in the table, the ninth stage had a space for two more nozzles so that we may take 150 nozzles for full admission. Consequently the arcs of admission will be found by multiplying the ratio of the number of nozzles in a stage to the number for total admission, by  $360^\circ$ . Thus for the first stage the arc of admission is

$$38 \times 360 \div 150 = 91.$$

If the arc of admission exceeds  $180^\circ$  then the unbalanced part will be the remainder found by subtracting from  $360^\circ$ . In the table there are set down the quarters of the angles of admission and their cosines.

The impulse for a stage may be found as follows: The shaft horse-power for the turbine is 8500 and the mechanical efficiency is 0.94 so that the turbine or gross horse-power is

$$8500 \div 0.94 = 9040,$$

and as there are 24 equal stages the horse-power per stage is

$$9040 \div 24 = 377.$$

The peripheral velocity is 375 feet per second, consequently the impulse is

$$377 \times 550 \div 375 = 553 \text{ pounds.}$$

This is the unbalanced impulse for the first five stages; for the other stages the impulse is in the proportion of the total arc of

admission to the unbalanced arc, that is for the sixth stage the impulse is

$$206 : 154 :: 553 : 413.$$

The last column of the table is found by multiplying the unbalanced impulses by the cosines of the preceding column. The total impulse is overestimated, perhaps to the extent of five per cent.

To estimate the importance of such unbalanced pressure, we may compare it with the weight of the rotor, which for the turbine in question is about 25 tons or 56,000 pounds; the unbalanced impulse is therefore only six or seven per cent.

**Conservation of Energy of Discharge.** — So far it has been assumed that the absolute velocity of discharge  $V_4$  from the blades of a stage of a pressure compound turbine is dissipated and the kinetic energy changed into heat. In the earlier stages of a turbine, when the number of nozzles increases from stage to stage, there must be an appreciable distance to allow the steam to spread and fill all the nozzles of the next set. Even in that case there may be an appreciable velocity of entrance to the nozzles, but when the admission is complete, or nearly so, there is no reason, aside from condition of construction, why there should be a large clearance between a wheel and the following nozzles. If the clearance is not more than half the width of the blade and if the steam does not spread much, nearly the entire velocity of discharge may be assumed as available for initial velocity of entrance to the nozzles.

With an initial velocity of entrance, the equation for the velocity of discharge without friction may be deduced from equation (2), page 15, by adding a term to represent the kinetic energy of discharge

$$\frac{V_1^2}{2g} = \frac{V_4^2}{2g} + E_1 - E_2 + p_1 v_1 - p_2 v_2. \quad \dots \quad (1)$$

For saturated steam this equation may be reduced to a form derived from equation (4), page 16,

$$\frac{AV_1^2}{2g} = \frac{AV_4^2}{2g} + x_1 r_1 + q_1 - x_2 r_2 - q_2. \quad \dots \quad (2)$$

It is perhaps best to affect the kinetic energy of discharge by a factor  $1 - y_d$  to allow for some dissipation of velocity; the total available heat and equivalent energy for the nozzle is then to be multiplied by a factor  $1 - y_n$  as usual. Arranging to use heat contents from the entropy table either for saturated or for superheated steam, we have

$$\frac{AV_1^2}{2g} = \left\{ \frac{AV_4^2}{2g} (1 - y_d) + C_1 - C_2 \right\} (1 - y_n) \quad . \quad . \quad (3)$$

which equation asserts that the heat equivalent of the available energy of discharge is to be added to the available heat from the entropy table.

If the blade angles  $\alpha$  and  $\beta$  are equal and there is no velocity of whirl at exit, we may recognize from Fig. 22, page 76, that

$$V_4 = \sqrt{1 - y_b} V_1 \sin \alpha. \quad . \quad . \quad . \quad (4)$$

where  $y_b$  is the factor for blade friction.

Inserting this value in equation (3) we have

$$\frac{AV_1^2}{2g} = \frac{AV_1^2}{2g} (1 - y_d) (1 - y_n) (1 - y_b) \sin^2 \alpha + (C_1 - C_2) (1 - y_n). \quad (5)$$

$$\therefore V_1 = \frac{223.7 \{ (C_1 - C_2) (1 - y_n) \}^{\frac{1}{2}}}{\{ 1 - (1 - y_b) (1 - y_d) (1 - y_n) \sin^2 \alpha \}^{\frac{1}{2}}} \quad . \quad . \quad . \quad (6)$$

in which  $y_b$ ,  $y_d$ ,  $y_n$  are factors to allow for degradation of kinetic energy into heat on account of blade friction, dissipation of velocity of discharge, and nozzle friction.

Suppose that  $\alpha = 20^\circ$ , and that each of the factors has the value 0.1, then the denominator is equal to

$$(1 - 0.9 \times 0.9 \times 0.9 \times 0.342^2)^{\frac{1}{2}} = 0.95$$

and consequently  $V_1$  is about

$$1 \div 0.95 = 1.05$$

times as large as when the energy of discharge is degraded into heat. The heat equivalent is proportional to the square of the velocity which gives a factor of about

$$1 \div 0.95^2 = 1.10,$$

which indicates a gain of about 10 per cent from the conservation of the energy of discharge. The heat factor per stage can be increased in a like proportion for a design, and the overall heat factor by a slightly less percentage; this last conclusion comes from the fact that a better economy and corresponding overall heat factor will call for a slightly smaller ratio for temperature distribution from the table on page 68. Finally the steam consumption will be reduced inversely proportional to the overall heat factor; that is, by something less than 10 per cent. For example, the overall heat factor for our design was taken (page 81) to be 0.73; the conservation of energy of discharge would increase this to about 0.80, and the steam consumption would be decreased from 11.85 to about 10.7 pounds per shaft horse-power per hour.

The above discussion gives a general idea of the advantage of the conservation of the energy of discharge; a design with that end in view should proceed systematically from the beginning with an estimate of the probable steam consumption or of the overall heat factor. The first stage does not get the advantage of the exit velocity from a preceding stage; on the contrary the velocity of approach will be taken as zero. Consequently it must be assigned a larger heat portion in compensation. Now the advantage of the conservation of energy of discharge is represented by the denominator of the right-hand number of equation (6).

$$1 - (1 - y_b)(1 - y_d)(1 - y_n) \sin^2 \alpha;$$

conversely the first stage which lacks that advantage may have a heat portion proportional to the reciprocal of this quantity. Otherwise the distribution of temperatures and pressures may proceed in the usual manner. The velocity of the jet  $V_1$  for all stages but the first will be computed by the equation

$$V_1 = \frac{223.7 \{ \text{heat} (1 - y_n) \}}{\{ 1 - (1 - y_b)(1 - y_n)(1 - y_d) \sin^2 \alpha \}^{\frac{1}{2}}}$$

where the heat is taken as the adiabatic assignment taking

account of the heat distribution factor of the first stage according to our usual form; in fact, the only difference is the introduction of the denominator to allow for the advantage of conservation; the first stage will commonly have the same velocity, which in that case need not be computed separately. Should it be desired to make a separate computation it will be made in the usual manner from the heat assignment, with no allowance for conservation.

Considered broadly the conservation of the energy of discharge gives the designer much greater latitude and in particular relieves him of the necessity of maintaining a small velocity of discharge; again, by shaping the entrance to the nozzles so that they may properly receive the steam from the preceding wheel, he is relieved from the condition that there should be no final velocity of whirl.

If there were no friction losses and if the energy of discharge could be completely conserved, then the turbine would come near to working on an equivalent to Rankine's cycle; the failure to do so can be charged to friction and partial dissipation of velocity. The nozzle angle may therefore be selected at discretion and will have no direct effect on efficiency. If the blades become inconveniently long, the nozzle angle may be increased so as to give the designer control over this feature; if, on the contrary, the blades become too short or the nozzles occupy too small a portion of the periphery, the nozzle angle can be reduced, but this last device is of more questionable value. The conditions for conservation are good when there is full peripheral admission for a pressure compound turbine, for the nozzles and blades follow closely; in fact conservation is apt to be attained whether or not the designer intends it.

As will be apparent after a discussion of the reaction turbine, an impulse turbine with one row of blades for each pressure stage, and with nozzles and blades following closely, is liable to approach the impulse and reaction type. It will be pointed out that the distinction is whether or not there is increase of velocity of the steam in passing through the blades, and this

will depend on whether the pressure is the same at exit as at entrance; should there be a fall of pressure in the blades, intentional or otherwise, then there will be an increase of velocity and a reaction. The chance for unintentional reaction will occur when there are a large number of pressure stages, having small drops of pressure; this condition is liable to occur in a low-pressure turbine or in some designs for the low-pressure end of a turbine.

The designer may take advantage of conservation of energy of discharge to reduce the peripheral velocity of a turbine without sacrificing the efficiency; there will be, of course, a larger friction loss with the higher steam velocities, which should be considered as a refinement of such a design.

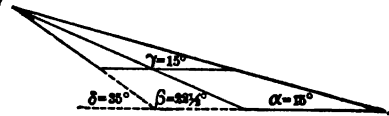


FIG. 36.

Such a design is presented in the turbine built by the Westinghouse Company and illustrated by Fig. 38 (facing page 114); some details of the blades and nozzles of this turbine have

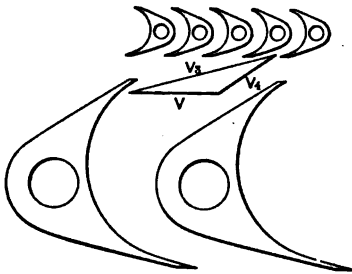


FIG. 37.

already been presented and a general description will be given later. The velocity of discharge from the nozzles is given as about 800 feet per second, while the peripheral velocity was 285 feet. The nozzle angle was about  $15^\circ$  as given by Fig. 33, page 103. Assuming a blade friction of  $\gamma = 0.1$  and that the blade exit

angle was about  $15^\circ$ , Fig. 36 was drawn thus determining  $35^\circ$  for the admission angle for the nozzles.

The arrangement of a set of blades and the following set of nozzles is shown by Fig. 37.

**Unequal Stages.** — In the discussion of the direct method of temperature and pressure distribution, it was stated that the available adiabatic heat could be divided into unequal portions

in Figs. 20 and 21 should occasion arise. A good illustration can be given if it be assumed that the energy of discharge is to be conserved, and if in addition the nozzle angles for the last two stages of the turbine design of page 81, are increased to  $30^\circ$ .

It was pointed out on page 111 that the heat that should be assigned to the first stage, which has not the advantage of initial velocity of steam approaching the nozzle, must be increased in order that the velocity of discharge  $V_1$  and also the peripheral velocity may be the same for this stage as for the following stages. Returning to the discussion on page 111 it appears that the factor for making this allowance is

$$\frac{1}{1 - (1 - \gamma_b)(1 - \gamma_d)(1 - \gamma_n) \sin^2 \alpha},$$

where  $\gamma_b$  and  $\gamma_n$  are the friction factors for blades and nozzles and  $\gamma_d$  is the factor to allow for dissipation of velocity. If these are all taken as 0.1 and if  $\alpha$  is  $20^\circ$  the factor becomes 1.093; this is the ratio of the heat portion for the first stage to the portions for succeeding stages with the same nozzle angle.

On the other hand, the velocity of discharge from the last two stages must be larger in order to give the same peripheral velocity as for the earlier stages; taking  $20^\circ$  for the nozzle angles for the earlier stages and  $30^\circ$  for the last two stages the jet velocities must be nearly in the ratio

$$\cos 20^\circ : \cos 30^\circ = 1.085,$$

the heat portions must be in the square of this ratio or 1.177. Summing up, we have one stage with a factor 1.093, 21 stages with the factor unity, and two stages with the factor 1.177, so that there will be the equivalent of

$$1.093 + 21 + 2 \times 1.177 = 24.45 \text{ stages.}$$

From a gauge pressure of 150 pounds to a vacuum of 28 inches there are 322.2 thermal units available for adiabatic action at entropy 1.56. The heat assigned to each of the intermediate stages will be

$$322.2 \div 24.45 = 13.18.$$

The heat portion for the first stage will be

$$1.093 \times 13.18 = 14.41$$

and for each of the last two stages

$$1.177 \times 13.18 = 15.51.$$

If it is desired to make computations for each fourth stage the heat assignments for the first and last intervals will be

$$\text{stages } 1 - 4; \quad 14.41 + 3 \times 13.18 = 53.95$$

$$\text{stages } 23 - 24; \quad 2 \times 15.51 + 2 \times 13.18 = 57.38$$

and for all the other stages the heat assignment will be

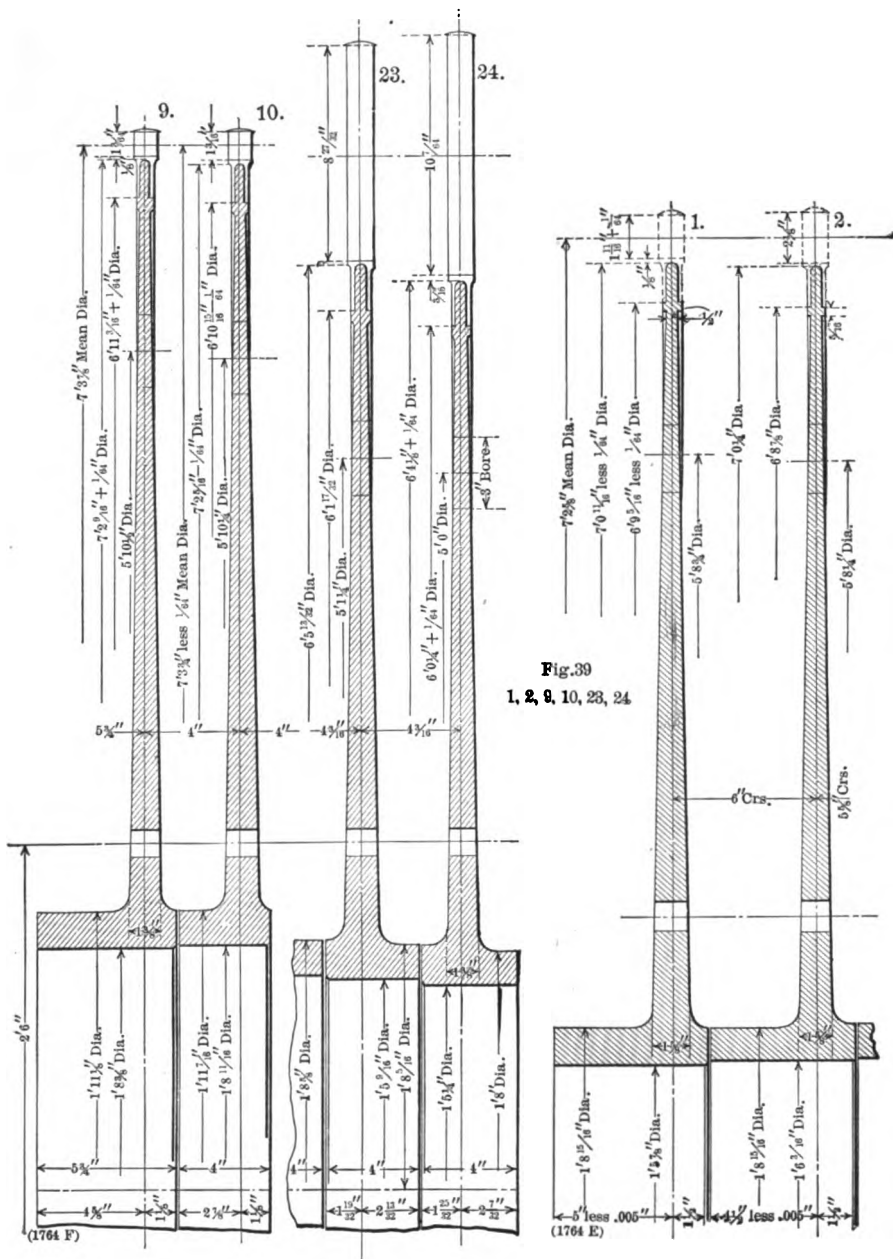
$$4 \times 13.18 = 52.72.$$

If the overall heat factor is taken as 0.80, as indicated on page 111, then the ratio for temperature distribution may be estimated as 1.04. A diagram is now to be drawn like Fig. 24, page 83, except that the stages are laid off unequal as stipulated. The intermediate temperatures and pressures are computed in the following table:

Stage.	Heat portions.	Factor for temperature distribution.	Heat allowance entropy 1.56.	Heat contents entropy 1.56.	Temperature.	Pressure.
0	.....	....	....	1193.3	366	164.8
4	53.96	1.034	55.8	1137.5	314	82.3
8	52.72	1.023	54.1	1083.4	266	39.2
12	52.72	1.002	52.8	1030.6	222	17.9
16	52.72	0.990	52.2	978.4	181	7.68
20	52.72	0.978	51.6	926.8	142	3.04
24	57.38	0.968	55.5	871.3	102	2.00

In this table the heat portions are multiplied by the temperature-distribution factors; the products are subtracted in succession from the heat contents at 366° entropy 1.56; after which the temperatures and pressures are found in the column for entropy 1.56.

**Westinghouse-Rateau Turbine.**—An illustrated description is given in *Engineering*, Jan. 13, 1911, of a turbine of the Rateau type built by the British Westinghouse Company for the Green-



**FIG. 39.**

wich Power Station, which is especially interesting because it gives constructive details with dimensions. Since the general dimensions computed for the design on page 81 correspond with this turbine, the details could be applied to that design.

The contract provided that the turbines, when supplied with steam at 180 pounds gauge and a vacuum of 27 inches, should develop 5000 KW. with a power factor of 0.85 so that the actual output was about 5890 KW. at 750 revolutions per minute. The builders guaranteed a consumption of 14.5 pounds per kilowatt hour (10.8 pounds per horse-power) with 180 pounds by the gauge and at 500° F., and with 28½ inches vacuum.

Fig. 38 gives a longitudinal section, and other details are given by Figs. 39 to 42; the blades and nozzles have already been illustrated in Figs. 30 to 33.

Of the twenty-four stages eight have partial admission and the others have full admission. The stages with full admission have the nozzles and blades following each other closely, so that there is nearly complete conservation of the energy of discharge from the wheels. The stages with partial admission are in three groups, for each of which groups the conservation is good; between the groups a chance is given for lateral spreading of the steam. Advantage is taken of the conservation to slow the turbine to a peripheral speed of 285 feet per second, without undue loss of efficiency. This reduction in peripheral speed, and consequently of revolutions per minute, is combined with unusually small angles for the nozzles and the exit from the blades at the high-pressure end of the turbine, and in consequence the passages through nozzles and between blades are much curved, while the velocities are high; it is questionable whether there is any advantage from the use of small angles in this combination.

The details of the wheels are given by Fig. 39; the blades and methods of securing them are given by Fig. 42, page 116. One of the diaphragms inside the nozzles is shown by Fig. 40 in elevations and section. A very important item in the construction



of the turbine is here illustrated, namely, that the casing, diaphragms, and all auxiliary parts are made in halves, so that the turbine may be opened for inspection or repair. For this purpose the diaphragm is parted along a diameter, and the joint is tongued and grooved to avoid leakage. Shaft leakage is checked by a ring, shown by Fig. 41, made in halves and dovetailed into the diaphragm; at the high-pressure end where the difference of pressure is large, the clearance at the shaft is as small as possible and the inside surface of the ring is serrated as shown at *A*, so that the white metal of which it is made may yield to the pressure of the shaft if necessary; when more clearance is allowed the surface is grooved as at *B*.

At the high-pressure end of the turbine where the admission is partial, the nozzles are formed as shown by Fig. 33, page 103; the groups are placed in recesses machined in the diaphragm. At the low-pressure end the nozzles become consecutive guides, as shown by Fig. 42, the exit angle being opened to  $30^\circ$ .

The glands to prevent leakage at the ends of the shaft are a combination of labyrinth packings and water seal. The construction and use of the labyrinth packing will be explained in Chapter X. Thus at the high-pressure end there is a labyrinth packing at *A*, Fig. 43, which reduces the pressure to 20-pounds pressure, and at *B* another reduces the pressure to that of the exhaust; the leakage from *A* is carried to a proper stage of the turbine, and from *B* to the condenser. The water seal has a paddle-wheel like a centrifugal pump, which throws the water to the outside of the chamber in which it runs and creates a pressure against which the atmosphere is unable to enter. The low-pressure end has only the water seal, but in

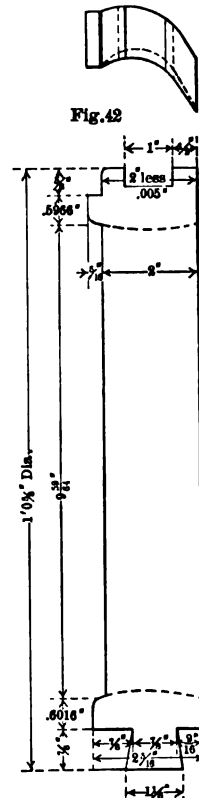


FIG. 42.

order to guard against entrance of air at starting there are special labyrinth packings.

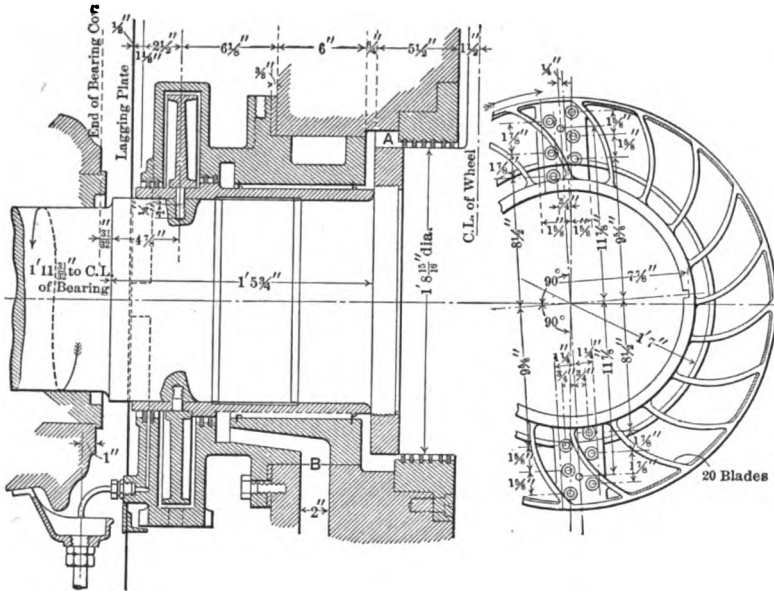


FIG. 43.

At the out-board end of the turbine shaft, as shown by Fig. 38, there is an effective thrust bearing with 12 collars and forced lubrication; this is more requisite as the high-pressure blades are shaped so as to give appreciable end thrust. The governor shaft is driven by spiral gears at this end of the shaft.

## CHAPTER VI

### VELOCITY COMPOUNDING

A VERY ready way of reducing the peripheral velocity of a steam-turbine is to expand the steam in a single set of nozzles from the initial steam-pressure to the back-pressure, as in the simple turbine, and let the steam act on a succession of moving and stationary vanes. For various reasons this method has been applied to turbines of moderate power only; but a combination with velocity compounding has been found well adapted to turbines of all powers and for various purposes.

**Velocities and Angles.** — Fig. 44 represents the arrangement of the blades and guides for using the jet; the blades are all set on the rim of the turbine wheel and the guides are attached to the turbine casing. The steam leaves the nozzles with the absolute velocity  $V_1$  from which the relative velocity  $V_2$ , with which it enters the first blades, can be determined by constructing the triangle  $V_1, V, V_2$ , making  $V$  equal to the peripheral velocity and  $\alpha$  equal to the nozzle angle. This construction determines also the blade angle  $\beta$  at entrance and  $\gamma$  in the figure is made equal to it. Neglecting friction the exit velocity  $V_3$  is made equal to  $V_2$ . The absolute exit velocity  $V_4$  is determined by the triangle  $V_3, V, V_4$ , with  $V$  equal to the peripheral velocity. The exit angle  $\alpha_1$  from the first guides is made equal to the entrance angle  $\delta$ .

In like manner the triangle  $V_1', V, V_2'$  determines  $V_2'$  and  $\beta_1$ ; and the triangle  $V_3', V, V_4'$  determines  $V_3'$  and  $\delta_1$ ; while the triangle  $V_1'', V, V_2''$  determines  $V_2''$  and the angle  $\beta_2$ .

The velocities and angles may be conveniently constructed by aid of the diagram, Fig. 45, in which the triangle  $V_1, V, V_2$  corresponds directly to the triangle so lettered in Fig. 44. The triangle  $V_2, V, V_1'$  takes the place of  $V_3, V, V_4$  in that figure,

$V_2$  being equal to  $V_3$  and  $V_1'$  equal to  $V_4$ . The remainder of the construction follows the same order. In the construction of

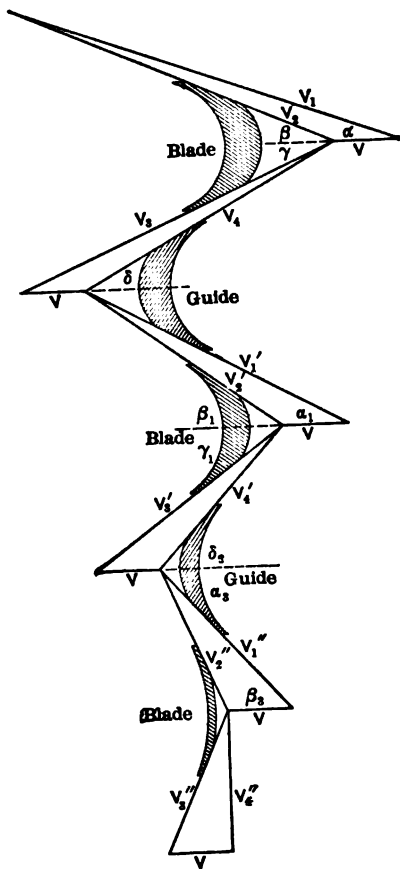


FIG. 44.

Figs. 45 and 46 the peripheral velocity  $V$  is taken as one-sixth of the initial velocity of whirl, so that the absolute exit velocity  $V_4''$  from the last set of blades is vertical and there is then no exit velocity of whirl.

It is to be borne in mind that the diagram of Fig. 45 is merely a convenient construction for finding velocities and angles; the student should familiarize himself with the more complete construction of Fig. 44 and learn to pass from one to the other with facility.

If all frictional and other losses can be ignored the efficiency of any turbine can be determined from the initial and final kinetic energies. In case of the diagrams, Figs. 44 and 45, the initial and final kinetic energies per pound of steam are

$$\frac{V_1^2}{2g} \text{ and } \frac{(V_4'')^2}{2g} = \frac{V_1^2 \sin^2 \alpha}{2g},$$

so that the energy applied to driving the turbine is

$$\frac{V_1^2}{2g} - \frac{V_1^2 \sin^2 \alpha}{2g} = \frac{V_1^2 \cos^2 \alpha}{2g}$$

and the efficiency is

$$\frac{V_1^2 \cos^2 \alpha}{2g} \div \frac{V_1^2}{2g} = \cos^2 \alpha. \quad . \quad . \quad . \quad (1)$$

This is the efficiency found on page 41 for the simple impulse turbine without axial thrust, and corresponds with the statement on page 73, that neglecting losses from friction and other causes, there is neither advantage nor disadvantage from compounding, so far as efficiency is concerned.

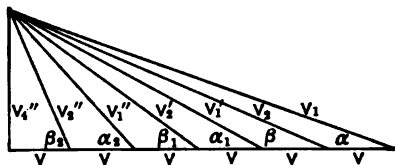


FIG. 45.

It is interesting and instructive to trace the application of energy to the several sets of blades; the sum of the works giving the basis for computing the efficiency, and giving the same results as equation (1).

In Fig. 44 the velocity of whirl at exit from the nozzle is

$$V_1 \cos \alpha,$$

and the velocity of whirl at exit from the first set of blades is

$$- V_4 \cos \delta = - \frac{4}{6} V_1 \cos \alpha;$$

consequently the work done on the vane is

$$\frac{w}{g} \left[ V_1 \cos \alpha - \left( - \frac{4}{6} V_1 \cos \alpha \right) \right] \frac{1}{6} V_1 \cos \alpha,$$

because the peripheral velocity of the blades is  $V_1$ , which is one-sixth of the initial velocity of whirl. This last expression reduces to

$$\frac{10}{36} \frac{w}{g} V_1^2 \cos^2 \alpha.$$

The second and third sets of blades receive the works

$$\frac{6}{36} \frac{w}{g} V_1^2 \cos^2 \alpha \quad \text{and} \quad \frac{2}{36} \frac{w}{g} V_1^2 \cos^2 \alpha,$$

so that the resultant work is

$$\frac{1}{2} \frac{w}{g} V_1^2 \cos^2 \alpha;$$

and this quantity divided by the initial kinetic energy gives the result set down in equation (1).

An instructive feature of this discussion is that the relation of the works done on the three sets of blades is

$$5, 3, 1.$$

A similar investigation will show that the distribution among four sets of blades is in the relation

$$7, 5, 3, 1.$$

The first figure in such a series is obtained by adding to the number of the sets of blades one less than that number. Five sets of blades would give the relation

$$9, 7, 5, 3, 1.$$

It will be noted that the last set of blades in a turbine which has three sets does only one-ninth of the work. The last set of four does only one-sixteenth of the work, and the last set of five does only one-twenty-fifth of the work.

The relations here set down will be change in practice by two considerations: in the first place friction of steam on the blades and guides will diminish the proportion for the last set; and on the other hand the distribution can be varied by varying the arrangement of angles from that shown by Fig. 45, and in particular the work of the last set of blades may be increased by decreasing its exit angle. Nevertheless, the actual distribution will not be very different from that set down.

The number of sets of blades seldom exceeds three; four sets are used in some marine turbines where the peripheral velocity is small and the diameter of the turbine case is large. There is no known case of the use of five sets of blades, but so many might have advantages for the backing turbine on shipboard.

For instruction, the treatment of the Figs. 44 and 45 has the advantage of simplicity; but it gives very flat and thin blades for the last set. A better result can be had by diminishing the exit angles for the guides, or for both guides and blades. If the blade angles are so reduced there is liable to be an axial thrust, but no more than can be taken easily by an end bearing. If there are only two sets of blades, the second set may not be too flat, but there may, nevertheless, be advantages from decreasing exit angles.

**Effect of Friction in Guides and Blades.** — An allowance for the influence of steam friction in guides and blades can be made

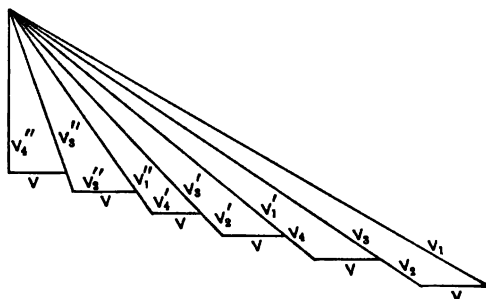


FIG. 46.

by an extension of the method given on page 76 for a single set of blades. Fig. 46 gives an application of that method to a turbine with three sets of blades, the entrance and exit angles being equal both for blades and guides. Taking  $V_1$  for the velocity of the jet and  $V$  for the peripheral velocity of the wheel, the triangle  $V_1, V, V_2$  gives, as before, the relative velocity  $V_2$  at entrance to the first set of blades. As the angles  $\beta$  and  $\gamma$  are equal, the line representing the exit velocity  $V_3$  coincides with  $V_2$ , but its length is less because friction reduces the velocity. To find the absolute velocity of discharge from the first wheel, the triangle  $V_3, V, V_4$  is drawn; this gives also the angle  $\delta$  for the entrance to the second set of guides. It must be remembered that the diagram gives the correct values for the velocities and angles, but the correct directions of the exit velocities  $V_3$  and  $V_4$

must be obtained by a diagram like Fig. 45. The process of construction can be repeated for the second set of blades by aid of the triangles  $V_1', V, V_2'$  and  $V_3', V, V_4$ , and is completed by drawing the triangles  $V_1'', V, V_2''$  and  $V_3'', V, V_4''$  for the third set of blades. The diagram was drawn by trial so that  $V_4''$  should be vertical, and that there is no exit velocity of whirl, a number of approximations being required for that purpose.

In order to obtain a clear diagram and to bring out the features of this method, the nozzle angle  $\alpha$  was taken  $30^\circ$ , and a large value was assigned to the friction factor, namely  $y = 0.19$ , so that

$$\sqrt{1-y} = 0.9,$$

this being the quantity by which each entrance velocity was multiplied to find the exit velocity; thus  $V_3$  is one-tenth less than  $V_2$  and  $V_1'$  is one-tenth less than  $V_4$ . The original diagram from which Fig. 46 was reduced was drawn to a convenient scale, so that measurements could be made with sufficient accuracy. Assuming the velocity of the jet to be 3500 feet per second, it was found that a peripheral velocity of 382 feet would complete the diagram, with no exit velocity of whirl. The initial velocity of whirl was

$$V_w = V_1 \cos \alpha = 3500 \cos 30^\circ = 3031;$$

it is to be noted that a frictionless diagram like Fig. 45 would give one-sixth of this, or 505 feet, for the peripheral velocity, which brings out clearly the feature that the effect of blade friction is to reduce the velocity for maximum efficiency.

The initial velocity of whirl may be measured on the diagram from the intersection of  $V_1$  and  $V$  to the vertical line  $V_4''$  produced, giving as before 3031 feet. In like manner the exit velocity of whirl can be measured from the intersection of  $V_4$  and  $V$  to the same vertical line; its length is 2002, and it has properly the negative sign. The velocities of whirl at entrance and exit, from the second and third sets of blades, can be measured in the same way. From the changes in the velocities of whirl and the velocity of the wheel  $V$ , the efficiency can be computed as follows:—

## COMPUTATION OF EFFICIENCY

	1	2	3
Velocity of whirl entrance . . . . .	3,031	1,802	806
Velocity of whirl exit . . . . .	-2,002	-896	.....
Retardation . . . . .	5,033	2,698	806
	<u>1,922,000</u>	<u>1,031,000</u>	<u>308,000</u>
Work of one pound of steam . . . . .	<u>g</u>	<u>g</u>	<u>g</u>
Ratio of works . . . . .	6.24	3.35	1.00
Total work of turbine . . . . .	<u>3,261,000</u>	.....	.....
	<u>g</u>		
Kinetic energy per pound of jet . . . . .		$\frac{3,500^2}{2g}$	
Efficiency . . . . .	$\frac{3,261,000}{g} \div \frac{3,500^2}{2g} = 0.532$		

The efficiency without friction as given by equation (1) is  $\cos^2 30^\circ = 0.75$ .

The excessive coefficient of friction unduly diminishes both the peripheral velocity and the efficiency, and also overaccentuates the undesirable distribution of work. There is also a notable reduction in the velocity of flow and, therefore, a considerable axial thrust, even though the blade angles are equal ( $\alpha = \beta$ , etc.).

The method given above, with modifications which will be obvious as occasion for them arises, can be applied to finding the efficiency for any impulse turbine, allowing for friction.

**Closure of Blade Angle.**—Whether or not the exit blade angles shall be diminished for a given turbine design will depend on the judgment of the designer. But in any case the exit angles of the guides should be diminished, because we can thereby avoid thin and flat blades for the last set, and at the same time somewhat improve the efficiency. It has been found that a convenient and conservative arrangement can be had if the exit guide angle be made equal to the preceding blade angle.

The construction for this arrangement is shown by Fig. 47, applied to two sets of blades. The triangles  $V_1, V, V_2$  and  $V_3, V, V_4$  are drawn as in Fig. 46, the angles  $\beta$  and  $\gamma$  for the first set of blades being equal. In this figure the factor

$$\sqrt{1 - \gamma} = 0.9,$$

in order to give a clear diagram and for the same reason  $\alpha$  is made  $30^\circ$ . The absolute exit velocity  $V_4$  from the first blades is affected by the multiplier given above and is laid off on the same line as  $V_2$  and  $V_3$ , so that  $\alpha_1 = \beta$ . The diagram is drawn by trial so that the final absolute velocity  $V_4'$  shall be vertical, giving no resultant velocity of whirl.

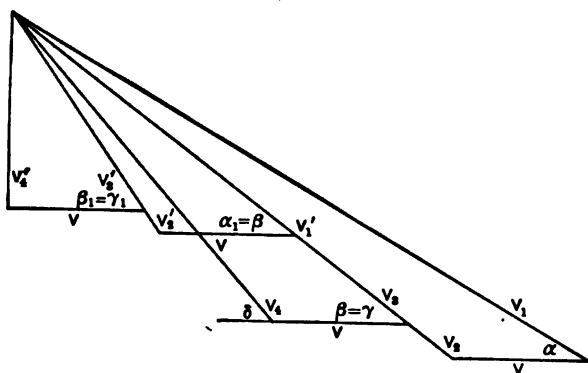


FIG. 47.

**Angles and Ratios.** — In preparing for design of turbines it is desirable that a series of diagrams for determining velocities and angles should be constructed with precision for several coefficients of friction and for two, three, and four sets of blades. All the velocities should be given in terms of the jet velocity  $V_1$ , so that after that velocity has been determined all the other velocities and the angles can be read from tabulated results. In both Fig. 46 and Fig. 47 the values of  $\gamma$  for all the velocities are taken the same (namely  $\gamma = 0.19$ ); but in practice it is customary to vary the factor for a given set of blades, depending on the velocities, it having been found that the factors are higher for large than for small velocities. This assignment of friction factors that increase with the velocity is in effect assuming that the resistance increases more rapidly than indicated by the square of the velocity.

Values for blade and guide friction factors have not been published and absolute values are but poorly known, to say

nothing of variations of factors. But the effect of variation of factors within moderate limits has only a secondary effect on the design, and students can select ratios of velocities and angles from the following tables; which have been derived from diagrams like Fig. 47, constructed with constant friction factors for the several velocities of the diagram.

### ANGLES AND RATIOS OF VELOCITIES

In these tables the blade angles are equal ( $\beta = \gamma$ ); the exit angle for a guide is made equal to the angle of the preceding blade ( $\alpha_1 = \gamma$ ).

Values are derived by construction and are consequently subject to slight errors.

The jet velocity  $V_1$  is taken as unity.

#### ONE WHEEL

*Nozzle Angle 20°*

Friction Coefficient ( $\gamma$ ) . . . . .	0.04	0.06	0.08	0.10	0.12	0.14	0.16
Blade angles (degrees) . . . . .	35½	35½	35½	35½	35½	35	35

*Velocities, Ratios to Jet Velocity  $V_1$*

Peripheral $V$ . . . . .	0.465	0.463	0.460	0.458	0.455	0.452	0.449
Wheel, entrance $V_2$ . . . . .	0.585	0.587	0.589	0.591	0.593	0.595	0.598
exit $V_3$ . . . . .	0.573	0.569	0.565	0.560	0.556	0.552	0.548
Efficiency . . . . .	0.874	0.869	0.865	0.860	0.855	0.850	0.845

#### TWO WHEELS

*Nozzle Angle 20°*

Friction Coefficient ( $\gamma$ ) . . . . .	0.04	0.06	0.08	0.10	0.12	0.14	0.16
Blade angles for all coefficients, 1st wheel, 25°-45'.							
Guide angles for all coefficients, Entrance, 36°-45', Exit, 25°-45'.							
Blade angles for all coefficients, 2d wheel, 45°-40'.							

*Velocities, Ratios to Jet Velocity  $V_1$*

Peripheral $V$ . . . . .	0.241	0.239	0.237	0.235	0.233	0.231	0.230
1st wheel, ent. $V_2$ . . . . .	0.788	0.788	0.788	0.788	0.788	0.788	0.788
exit $V_3$ . . . . .	0.772	0.764	0.756	0.748	0.739	0.730	0.722
Guide, ent. $V_4$ . . . . .	0.553	0.548	0.544	0.538	0.533	0.529	0.525
exit $V_1$ . . . . .	0.542	0.530	0.520	0.510	0.499	0.490	0.481
2d wheel, ent. $V_2$ . . . . .	0.341	0.333	0.324	0.310	0.308	0.300	0.293
exit $V_3$ . . . . .	0.334	0.323	0.312	0.301	0.290	0.280	0.269
Efficiency . . . . .	0.90	0.89	0.88	0.86	0.85	0.84	0.83

## THREE WHEELS

Nozzle Angle  $20^\circ$ 

Friction Coefficient ( $\gamma$ ) . . . . .	0.04	0.06	0.08	0.10	0.12	0.14	0.16
Blade angles							
1st wheel . . . . .	23-40	23-40	23-30	23-30	23-20	23-10	23-0
1st guide, ent. . . . .	28-10	28-50	28-30	28-20	28-0	27-40	27-30
exit . . . . .	23-40	23-40	23-30	23-30	23-20	23-10	23-0
Blade angles							
2d wheel . . . . .	30-50	30-30	30-10	29-50	29-30	29-20	29-0
2d guide, ent. . . . .	43-0	42-30	41-60	41-20	40-50	40-20	39-40
exit . . . . .	30-50	30-30	30-10	29-50	29-40	29-20	29-0
Blade angle							
3d wheel . . . . .	49-40	49-10	48-40	48-10	47-40	47-10	46-40

Velocities, Ratios to Jet Velocity  $V_1$ 

Peripheral V . . . . .	0.162	0.158	0.153	0.148	0.143	0.138	0.133
1st wheel, ent. $V_2$ . . . . .	0.849	0.853	0.858	0.862	0.867	0.871	0.875
exit $V_3$ . . . . .	0.832	0.830	0.828	0.823	0.817	0.807	0.793
1st guide, ent. $V_4$ . . . . .	0.686	0.689	0.690	0.689	0.687	0.682	0.673
exit $V_1'$ . . . . .	0.671	0.667	0.661	0.653	0.643	0.630	0.617
2d wheel, ent. $V_5$ . . . . .	0.530	0.529	0.526	0.522	0.515	0.506	0.497
exit $V_3'$ . . . . .	0.518	0.514	0.507	0.497	0.485	0.470	0.454
2d guide, ent. $V_6$ . . . . .	0.392	0.388	0.384	0.377	0.368	0.357	0.345
exit $V_1''$ . . . . .	0.379	0.376	0.369	0.360	0.348	0.335	0.320
3d wheel, ent. $V_7''$ . . . . .	0.255	0.251	0.244	0.237	0.228	0.220	0.211
exit $V_3''$ . . . . .	0.249	0.240	0.231	0.222	0.213	0.204	0.195
Efficiency . . . . .	0.90	0.87	0.83	0.80	0.77	0.74	0.70

## THREE WHEELS

Nozzle Angle  $25^\circ$ 

Friction Coefficient ( $\gamma$ ) . . . . .	0.04	0.06	0.08	0.10	0.12	0.14	0.16
Blade Angles							
1st wheel . . . . .	29-40	29-30	29-20	29-10	29-0	28-50	28-40
1st guide angles							
ent. . . . .	36-20	35-50	35-30	35-10	34-50	34-20	34-0
exit . . . . .	29-40	29-30	29-20	29-10	29-0	28-50	28-40
Blade angles							
2d wheel . . . . .	38-0	37-40	36-20	36-50	36-30	36-10	35-50
2d guide angles							
ent. . . . .	51-30	51-0	50-20	49-40	49-0	48-30	47-50
exit . . . . .	38-0	37-40	37-20	36-50	36-30	36-10	35-50
Blade angles							
3d wheel . . . . .	57-0	56-30	56-0	55-40	55-10	54-50	54-20

Velocities, Ratios to Jet Velocity  $V_1$ 

Peripheral V . . . . .	0.163	0.158	0.153	0.148	0.144	0.139	0.134
1st wheel, ent. $V_2$ . . . . .	0.857	0.860	0.864	0.868	0.871	0.875	0.878
exit $V_3$ . . . . .	0.838	0.833	0.828	0.823	0.818	0.814	0.809
1st guide, ent. $V_4$ . . . . .	0.702	0.700	0.699	0.697	0.696	0.694	0.693
exit $V_1'$ . . . . .	0.688	0.679	0.670	0.661	0.653	0.644	0.635
2d wheel, ent. $V_5$ . . . . .	0.553	0.548	0.542	0.536	0.532	0.526	0.521
exit $V_3'$ . . . . .	0.542	0.531	0.521	0.510	0.499	0.489	0.478
2d guide, ent. $V_6$ . . . . .	0.425	0.417	0.409	0.401	0.393	0.385	0.377
exit $V_1''$ . . . . .	0.417	0.405	0.393	0.381	0.369	0.357	0.345
3d wheel, ent. $V_7''$ . . . . .	0.306	0.297	0.287	0.277	0.267	0.257	0.248
exit $V_3''$ . . . . .	0.298	0.287	0.275	0.263	0.252	0.240	0.228
Efficiency . . . . .	0.87	0.84	0.81	0.78	0.75	0.72	0.69

## FOUR WHEELS

Nozzle Angle  $20^\circ$ 

Friction Coefficient ( $\gamma$ )	0.04	0.06	0.08	0.10	0.12	0.14	0.16
Blade angles							
1st wheel . . . . .	22-30'	22-30'	22-20'	22-20'	22-20'	22-10'	22-10'
1st guide angles							
ent. . . . .	25-50	25-40	25-30	25-20	25-10	25-0	24-50
exit . . . . .	22-30	22-30	22-20	22-20	22-20	22-10	22-10
Blade angles							
2d wheel . . . . .	26-40	26-30	26-20	26-10	26-0	25-40	25-20
2d guide angles							
ent. . . . .	32-50	32-20	31-50	31-20	31-0	30-30	30-0
exit . . . . .	26-40	26-30	26-20	26-10	26-0	25-40	25-20
Blade angles							
3d wheel . . . . .	34-50	34-20	33-50	33-30	33-00	32-40	32-10
3d guide angles							
ent. . . . .	48-20	47-10	46-10	45-20	44-20	43-30	42-30
exit . . . . .	34-50	34-20	33-50	33-30	33-00	32-40	32-10
Blade angles							
4th wheel . . . . .	53-40	53-0	52-20	51-30	50-50	50-10	49-30

Velocities, Ratios to Jet Velocity  $V_1$ 

Peripheral $V$ . . . . .	0.121	0.117	0.112	0.108	0.103	0.098	0.093
1st wheel, ent. $V_2$ . . . . .	0.888	0.893	0.897	0.901	0.905	0.910	0.915
exit $V_3$ . . . . .	0.870	0.865	0.860	0.855	0.850	0.845	0.840
1st guide, ent. $V_4$ . . . . .	0.762	0.760	0.758	0.757	0.755	0.753	0.752
exit $V_1$ . . . . .	0.742	0.735	0.727	0.719	0.710	0.700	0.689
2d wheel, ent. $V_2$ . . . . .	0.635	0.630	0.625	0.620	0.615	0.610	0.606
exit $V_3$ . . . . .	0.619	0.611	0.602	0.591	0.580	0.567	0.544
2d guide, ent. $V_4'$ . . . . .	0.515	0.509	0.503	0.496	0.489	0.481	0.473
exit $V_1''$ . . . . .	0.504	0.493	0.482	0.470	0.458	0.446	0.433
3d wheel, ent. $V_2''$ . . . . .	0.403	0.394	0.386	0.379	0.369	0.360	0.351
exit $V_3''$ . . . . .	0.394	0.382	0.370	0.358	0.346	0.334	0.322
3d guide, ent. $V_4''$ . . . . .	0.303	0.294	0.285	0.275	0.266	0.256	0.247
exit $V_1'''$ . . . . .	0.295	0.283	0.272	0.261	0.249	0.238	0.227
4th wheel, ent. $V_2'''$ . . . . .	0.208	0.199	0.191	0.182	0.173	0.164	0.155
exit $V_3'''$ . . . . .	0.203	0.193	0.183	0.172	0.162	0.152	0.141
Efficiency . . . . .	0.89	0.85	0.81	0.77	0.73	0.70	0.66

**Construction of Blades and Guides.** — If the blades are given equal entrance and exit angles they may be constructed as shown by Fig. 26, page 89; if either blades or guides have unequal angles the construction shown by Fig. 48 can be used. Beginning at the entrance  $o$  the line  $ol$  may be drawn vertically and the line  $om$  may be drawn making the angle

$$lom = \frac{1}{2}(\beta - \gamma).$$

From  $o$  and  $m$  the lines  $on$  and  $mn$  may be drawn, making the angles  $\beta$  and  $\gamma$  with the horizontal, and the lines  $op$  and  $mp$  may be drawn perpendicular to them, intersecting in the center

$p$ , from which the arc  $om$  may be drawn for the face of the blade (or guide). The peripheral width of the blade edge can be laid

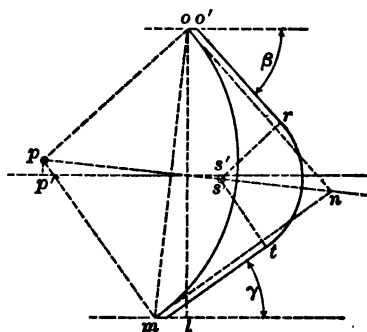


FIG. 48.

off at  $oo'$ , and the straight part of the back can be drawn parallel to  $on$ ; in like manner the straight part at the exit may be drawn parallel to  $mn$ . From  $p$  the line  $pp'$  is drawn perpendicularly to the medial line  $p's'$ , and the pitch is laid off equal to  $p's'$  to locate the center  $s'$ , the center from which the round of the back  $rt$  is drawn. Strictly, the

thickness of the edge measured perpendicular to  $on$  and to  $mn$  should be equal at entrance and exit, but as the angle  $lom$  is not large the peripheral dimensions can be made equal if that is found to be more convenient.

It will appear that as the clear passage equal to

$$po - sr$$

increases as the blade and guide angles increase, so also should the blank spaces  $oo'$ , in order that the condition discussed on page 95 shall obtain; such increase is not desirable and we may, therefore, maintain the distance  $oo'$  constant, or we may choose a thickness of edge perpendicular to  $po$ , and maintain that dimension constant for all the blades and guides of a velocity compound turbine.

**Design of a Velocity Compound Turbine.** — Let it be required to design an impulse turbine having three velocity stages with the following conditions:

Shaft horse-power . . . . .	475
Pressure by the gauge, pounds . . . . .	138.2
Vacuum, inches . . . . .	26
Angle of nozzle . . . . .	20°
Friction coefficient for nozzles $\gamma_n$ . . . . .	0.10
Friction coefficient for blades and guides $\gamma_b$ . . . . .	0.16
Heat losses due to rotation, etc. . . . .	0.90
Mechanical efficiency . . . . .	0.90

This type of turbine is unusual and the conditions have been taken largely at random, the purpose being mainly to illustrate the features of the design, some of which are found in pressure and velocity compound turbines. In particular the value of the friction factor  $\gamma_b$  is probably too small.

The pressures being the same as for the design on page 44, we shall have the same preliminary calculations, which will be summarized here. The temperatures corresponding to the assigned gauge pressure and vacuum are  $360^\circ$  and  $126^\circ$ , and the entropy table is entered at entropy 1.56. The absolute entrance pressure for the nozzle is 152.9 pounds, so that the throat pressure is

$$0.58 \times 152.9 = 88.7 \text{ pounds,}$$

which corresponds to the temperature  $319^\circ$ .

The available heats for adiabatic expansion to throat and exit are

$$1187.1 - 1143.2 = 43.9 \text{ B.T.U.}$$

$$1187.1 - 904.9 = 282.2 \text{ B.T.U.}$$

No allowance for friction will be made in computing velocity at the throat, but for the exit we will take  $\gamma = 0.1$  as given in the conditions, so that the velocities are

$$\text{Throat } 223.7 \sqrt{43.9} = 1483;$$

$$\text{Exit } 223.7 \sqrt{282.2 \times 0.9} = 3566.$$

The thermal efficiency for Rankine's cycle is

$$\sqrt{e} = 1 - \frac{C_3 - q_3}{C_1 - q_3} = 1 - \frac{904.9 - 94.0}{1187.1 - 94.0} = 0.258.$$

The efficiency of the action of the steam in the blades and guides, as affected by friction, from the table on page 130, is 0.70. Allowing for the loss from disc friction, etc. (factor 0.9) and for the mechanical efficiency 0.9, the general efficiency of the turbine is

$$\begin{array}{cccccc} \text{Nozzle} & \text{Thermal} & \text{Blade} & \text{Disc, etc.} & \text{Heat} & \\ 0.9 \times 0.258 \times 0.70 \times 0.9 \times 0.9 = 0.132, \end{array}$$

the first factor 0.9 being for the nozzle, which is given a friction factor  $\gamma = 0.1$ .

Now one horse-power is equivalent to 42.42 B.T.U. per minute, so that with an efficiency of 0.132 the thermal units required per brake horse-power per minute will be

$$42.42 \div 0.132 = 321 \text{ B.T.U.}$$

To vaporize a pound of steam there are required

$$x_1 r_1 + q_1 - q_2 = 1187.1 - 94.0 = 1093.1 \text{ B.T.U.,}$$

so that the steam per brake horse-power per hour is

$$321 \times 60 \div 1093.1 = 17.6 \text{ pounds.}$$

The total hourly consumption is, therefore, .

$$17.6 \times 475 = 8360 \text{ pounds nearly.}$$

The specific volumes as determined on pages 24 and 25

$$\text{Throat } 4.75; \quad \text{Exit } 142.8;$$

consequently the total nozzle areas are

$$\text{Throat } \frac{4.75 \times 8360 \times 144}{3600 \times 1483} = 1.07 \text{ square inches;}$$

$$\text{Exit } \frac{142.8 \times 8360 \times 144}{3600 \times 3566} = 13.4 \text{ square inches.}$$

Assuming that there are to be twelve nozzles for the normal load, the area for a single nozzle will be

$$\text{Throat } 1.07 \div 12 = 0.089 \quad \text{Exit } 13.4 \div 12 = 1.12.$$

If the throat section is circular its diameter will be 0.336 of an inch; and if the exit is one inch long measured radially its net width will be 1.12 inches.

Before proceeding with the design of the nozzles, blades, and guides we should determine the diameter of the pitch surface, so that the number of blades and their pitch can be properly assigned. The peripheral velocity is

$$3566 \times 0.133 = 474 \text{ feet per second,}$$

the factor for peripheral velocity, 0.133, being taken from the table on page 130 with  $\gamma$  equal to 0.16. Assuming 3000 revolutions per minute, the perimeter of the wheel becomes

$$474 \times 60 \div 3000 = 9.48 \text{ feet or } 113.8 \text{ inches.}$$

The corresponding pitch diameter is 3.02 feet or 36.21 inches. These dimensions will be found to give a convenient design; should the first assignment of revolutions per minute give undesirable proportions another would be chosen.

The width of blades and guides measured axially will be determined from experience or by trial. In this case a width of 1.5 inches will be found convenient with a pitch 0.6 of the width. If we take 128 blades and guides for each set the pitch will be

$$113.8 \div 128 = 0.889 \text{ of an inch,}$$

which is a trifle less than the proportion assigned.

Taking the peripheral width of the blade edge 0.03 of an inch, the effective pitch will be 0.859 of an inch. Now the peripheral width of the nozzle will be

$$1.116 \div \sin \alpha = 1.116 \div 0.342 = 3.263 \text{ inches,}$$

so that there will be

$$3.263 \div 0.859 = 3.80 \text{ blades per nozzle,}$$

which is a fair proportion. In order that the condition discussed on page 95 may obtain, the blank peripheral space between nozzles will be

$$3.80 \times 0.03 = 0.114 \text{ of an inch.}$$

The pitch of the nozzles will therefore be

$$3.263 + 0.114 = 3.377 \text{ inches.}$$

It will depend on the construction of the turbine, the method of governing, and the discretion of the designer whether the twelve nozzles shall be arranged in one group, or divided into two or more groups.

**Nozzles, Blades, and Guides.** — The arrangement of the nozzles, blades, and guides for the design just computed is shown by Fig. 49, which is a developed cylindrical section by the pitch surface.

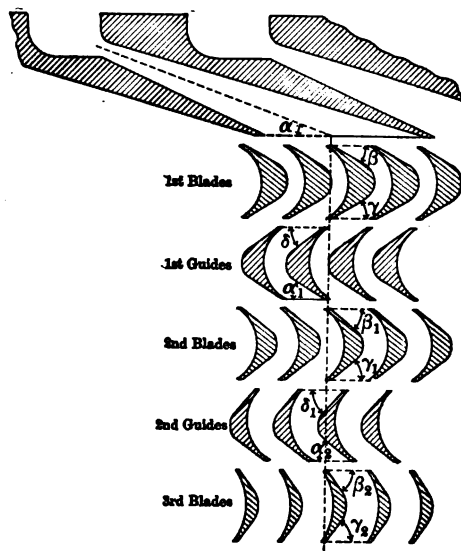


FIG. 49.

A rounded bowl leads from the steam space down to the throat of the nozzle, which, as stated in the design, is circular. The nozzle gradually changes from the circular throat to a rectangular exit, or more properly such a form would be used if the diameter of the wheel were very large compared with the radial length of the orifice. As a matter of fact, the section of the nozzle by the lower surface of the casting from which it is worked is a sector of an annular ring, as shown by Fig. 50.

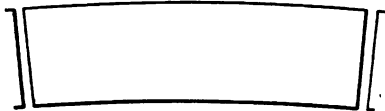


FIG. 50.

The nozzle is first drilled through in the proper position and direction, and is brought to the form at exit by filing. To facilitate this operation a steel former is made, which has the exact shape of the

interior of the nozzle when finished. After the interior is brought approximately to shape, this former surfaced with paint is forced in and high spots are marked which are then reduced by filing. The shape of the former is complex, having a circular section at right angles to the axis of the nozzle at the throat, and at the exit the section made by a plane at an angle of  $20^\circ$  to the axis must have a section like Fig. 50. The construction should be made to double scale or larger, and the dimensions determined accurately for the information of the machinist who makes the former.

The nozzles thus made by filing must be straight, as indicated by Fig. 50, and the angles are smaller at one side and larger at the other than the assigned value; in the figure they are  $17^\circ$  and  $25^\circ$  instead of  $20^\circ$ , a larger variation than would occur in practice when such nozzles are used for pressure and velocity-compound turbines. The nozzles follow in close succession and the mean angle of the steam discharged from them is  $20^\circ$  as assigned. The throats are sometimes rectangular in section, or in some cases are oblong, made by drilling two holes close together and joining them by straight sides.

The blades have equal angles and are constructed as shown by Fig. 26, page 89; the guides are given smaller angles at exit than at entrance, and are constructed as shown by Fig. 49. The peripheral widths of the edges are all 0.03 of an inch; this construction thickens up the last set of blades but gives them blunt edges.

On account of the difficulty and expense of making filed nozzles, there is a tendency to substitute reamed nozzles similar to that shown on page 20 for the de Laval turbine. Such reamed nozzles are sometimes given the form of a straight cone, which thus have smaller and larger angles than those assigned, similar to the nozzle of Fig. 49. The exit orifices at the inner surface of the plate in which the nozzles are bored are thus elliptical, like those shown by Fig. 51. Taking the same exit areas as in the preceding design, the

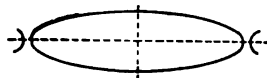


FIG. 51.

diameter will be 1.19 inches. In order to give a definite construction, it will be assumed that the nozzle, like that shown on page 20, is prolonged by a cylindrical shell; the major diameter of the ellipse will therefore be

$$1.19 \div \sin 20^\circ = 3.48 \text{ inches.}$$

If the peripheral space from one ellipse to another is 0.12 of an inch, the pitch will be 3.60 inches, and there will be

$$3.60 \div 0.889 = 4.05$$

blades for each nozzle.

**Lengths of Blades and Guides.** — The radial length of the first set of blades should be at least equal to the radial dimension of the nozzles. The lengths of succeeding guides and blades will increase, on account of friction of the steam and on account of reduction of exit angles of either guides or blades.

In order to make this matter clear consider the diagram of Fig. 44, page 122, which was drawn with equal angles for blades and guides, and without allowance for steam friction. In this diagram the peripheral widths of the edges of both blades and guides are equal. Such a construction calls for equal radial lengths of all blades and guides. It will be noted in this figure, and more explicitly in Fig. 45, that the velocity of flow is constant. Now the relative velocity of the steam entering the first set of blades can be resolved into two components, parallel to the direction of motion of the wheel and perpendicular to that direction; the latter component in Fig. 45 is equal to the velocity of flow. Turning to Fig. 49 the volume of steam entering between two blades may be found by taking the continued product of the axial blade-length, the clear peripheral distance between blade and blade, and the velocity of flow; the clear distance is equal to the pitch minus the peripheral width of the blade edge. To show that this is true consider that the volume can be found by taking the continued product of the blade-length, the net distance from blade to blade, and the relative velocity. But the net distance is the clear distance multiplied

by the sine of the blade angle  $\beta$ , and the relative velocity is the velocity of flow divided by the sine of  $\beta$ ; consequently the continued products are equal. If there were no friction there would be no change in volume of the steam, and consequently the continued product of the length, clear distance, and velocity of flow would be constant; but in Fig. 45 the conditions make the clear distance and the velocity of flow constant, and the blade lengths must also be constant. Of the several influences affecting the lengths of blades and guides, the most effective is reducing the exit angle; next comes the reduction of velocity of the steam on account of friction, and lastly the drying and consequent increase of specific volume of the steam.

As already explained on page 95 the following relation should exist when the steam passes from the nozzles into the first set of blades: namely, the peripheral distance from one nozzle to the next, measured along the edge of the dividing plate or member, should be equal to the peripheral width of the blade edge multiplied by the number of blades per nozzle. Thus, in the design just computed, that distance is

$$0.03 \times 3.80 = 0.114$$

as shown on page 135. The proper blade length will then be equal to the radial length of the nozzle at exit. This relation comes from the fact that the velocity of flow from the nozzle is also the velocity of flow into the blades; this is true in any case whether or not the exit angles are reduced and whether or not account is taken of friction. The velocity of flow being the same the clear space, measured on the periphery, for a nozzle must be equal to the sum of the clear spaces between the corresponding number of blades. The extended discussion of the relations of clear spaces and blank spaces is given, so that the allowance for friction may be the better understood.

Having the length of the first set of blades at entrance, the other lengths of blades and guides may be found by making them directly proportional to the specific volumes, and inversely

proportional to the velocities and the net distances; the velocities are relative for blades and absolute for guides. The computation can be shown by an example.

**Design of Velocity Compound Turbines** (*continued*). — The remainder of the design begun on page 132 consists in the determination of the blade and guide lengths at entrance and exit. This work may be tabulated as follows:

	Velocity.	Heat equivalent.	Dif.	Heat contents.	Volumes.	Angles.	Sines.	Length.
Nozzle . . . . .	$V_1$	3566	254.0	933.1	142.8	20-0	0.3420	1.00
Peripheral velocity . . . . .	$V$	474						
1st wheel, entrance . . . . .	$V_2$	3120	194.5	933.1	142.8	23-0	0.3907	1.00
exit . . . . .	$V_2'$	2830	160.0	967.6	148.7	do.	do.	1.15
1st guide, entrance . . . . .	$V_3$	2400	115.1	967.6	148.7	27-30	0.4617	1.15
exit . . . . .	$V_3'$	2200	96.7	986.0	151.9	23-0	0.3907	1.51
2d wheel, entrance . . . . .	$V_2''$	1773	62.8	986.0	151.9	29-0	0.4848	1.51
exit . . . . .	$V_2'''$	1620	52.2	996.6	153.7	do.	do.	1.67
2d guide, entrance . . . . .	$V_4$	1230	30.2	996.6	153.7	39-40	0.6383	1.67
exit . . . . .	$V_4''$	1142	26.0	1000.8	154.4	29-0	0.4848	2.40
3d wheel, entrance . . . . .	$V_3'''$	752	11.2	1000.8	154.4	46-40	0.7274	2.40
exit . . . . .	$V_3''''$	696	9.7	1002.3	154.6	do.	do.	2.61

The nozzle velocity is copied down from page 133, and the other velocities are found by multiplying this velocity by the ratios found in the table on page 130 for three wheels, nozzle angle  $20^\circ$  and friction factor  $\gamma = 0.16$ . The heat equivalents are computed by the equation

$$\text{Heat equivalent} = V^2 \div 2g \times 778 = 0.00001998 V^2,$$

which is the converse of equation (12), page 22. The heat equivalent for the nozzle velocity  $V_1$  is, of course

$$0.9 \times 282.2 = 254.0,$$

derived from page 133 and is introduced in the table as a convenience. The other heat equivalents are readily computed by aid of a table of squares.

In the first set of blades the velocity is reduced by friction from 3120 to 2830 feet per second; the heat equivalents are 194.5 and 160.0, so that the heat resulting from the loss of kinetic energy is 34.5 B.T.U.; in like manner the remainder of the column of differences is filled out.

The initial heat contents at 360° was 1187.1 B.T.U., and of this 254.0 B.T.U. were changed into kinetic energy at exit from the nozzles, leaving 933.1 in the steam; the corresponding specific volume was found by interpolation in the entropy tables at 126°, as explained on page 25. The steam entered the first blades with this specific volume, but the effect of friction was to dry the steam and increase its specific volume; the heat contents at exit from the blades was

$$933.1 + 34.65 = 967.6 \text{ B.T.U.},$$

and the specific volume corresponding was found by interpolation in the entropy table at 126° to be 148.7 cubic feet. In like manner the other specific volumes were determined. The specific volume is, of course, the same at exit from a wheel and entrance to the next guides, and at exit from a set of guides and entrance to the next blades; the table can be somewhat contracted to advantage, especially when a turbine with several pressure stages is computed, as in the next chapter.

The blade and guide angles are copied down from the table on page 130 and the natural sines are entered with them.

The net distance from the back of a blade or guide to the face of the next blade or guide, as is apparent from Fig. 50, is equal to the clear space measured along the periphery multiplied by the sine of the angle; and since in this design the peripheral width of the blade and guide edge is constant (0.03 of an inch) the net distance is proportional to the sine of the angle. Consequently the lengths of blades and guides are proportional to the specific volumes, and inversely proportional to the velocity and to the sines of the angles. The same relation applies to the radial width of the nozzles, because the blank peripheral spaces have been properly apportioned to that end. The radial length of the

nozzle was taken to be one inch, consequently the first blade length at entrance is

$$\frac{142.8}{142.8} \times \frac{3566}{3120} \times \frac{0.3420}{0.3907} = 1,$$

as should be the case.

The first blade length at exit is

$$\frac{148.7}{142.8} \times \frac{3566}{2830} \times \frac{0.3420}{0.3907} = 1.15.$$

The first guide length at entrance is

$$\frac{148.7}{142.8} \times \frac{3566}{2400} \times \frac{0.3420}{0.4617} = 1.15.$$

The first guide length at exit is

$$\frac{151.9}{142.8} \times \frac{3566}{2200} \times \frac{0.3420}{0.3907} = 1.51.$$

The other lengths of blades and guides are found in like manner and are recorded in the table on page 140. The length of a blade at exit is the same as the length of the next guide at entrance, and a similar condition holds for the entrance to the next set of blades; this condition was, of course, fulfilled in this design; other methods of laying out blades and guides are liable to give inequality at such places. The entrance length of a blade or guide should not be less than that of the exit from the preceding guide or blade, and if necessary any length may be arbitrarily increased to avoid such a contingency.

Some designers give a constant normal thickness to blades and guides, and this condition is likely to lead to the kind of inequality just mentioned. Suppose that the thickness in the design in question were made 0.02 of an inch, other conditions being unchanged. The net distance at the entrance to the first blades, from back to face, will in that case be the pitch multiplied by the sine of the angle minus the thickness, or

$$\text{pitch} \times \sin \beta - 0.02,$$

and a similar relation holds for the exit from these blades and for the following guides and blades, taking the proper angle instead of  $\beta$ . At entrance this gives

$$0.889 \sin 23^\circ - 0.02 = 0.327 \text{ inch.}$$

Now the specific volume at entrance to the first blades is the same as at the exit from the nozzles, and consequently the areas are inversely proportional to the velocities  $V_1$  and  $V_2$ , *i.e.*

$$\text{Area} = 0.294 \times \frac{3566}{3120} = 0.336.$$

Consequently the blade height at entrance will be

$$0.336 \div 0.327 = 1.03.$$

At exit the specific volume is greater than at entrance, on account of the effect of friction, *i.e.*, 148.7 cubic feet instead of 142.8; there must be a corresponding increase in the net area. Allowing for reduction in velocity and increase in specific volume the area at exit from the blades must be

$$0.294 \times \frac{3566}{2830} \times \frac{148.7}{142.8} = 0.386.$$

The net distance is

$$0.889 \sin 23^\circ - 0.2 = 0.327 \text{ of an inch}$$

as previously calculated; this recomputation is inverted because for the guides the angle changes, as from  $27^\circ-30'$  to  $23^\circ$  for the first set. The blade length at exit is therefore

$$0.386 \div 0.327 = 1.18 \text{ inches.}$$

The blade heights computed by the two methods are as follows:

	First blades.		First guides.		Second blades.		Second guides.		Third blade.	
	Ent.	Exit.	Ent.	Exit.	Ent.	Exit.	Ent.	Exit.	Ent.	Exit.
Constant peripheral width .	1.00	1.15	1.15	1.51	1.51	1.67	1.67	2.40	2.40	2.61
Constant thickness . . . .	1.03	1.18	1.17	1.55	1.53	1.70	1.68	2.39	2.40	2.61

The choice of thickness of edge makes the peripheral width slightly greater than 0.03 of an inch for the first set of blades, and consequently calls for a little greater blade length; this will hardly give the designer much trouble. But the discrepancy of

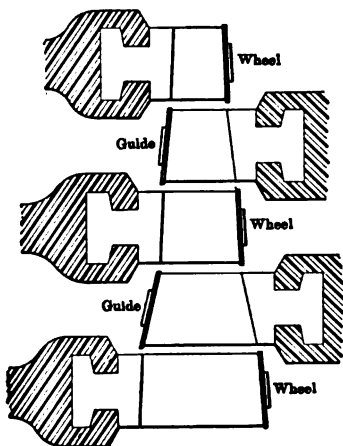


FIG. 52.

a smaller entrance height for the second blades than for the exit of the preceding guides would be a difficulty, if the designer did not exercise his discretion of increasing it to the same amount. But after all the discrepancies are of minor importance.

**Blades and Guides.** — The lengths of the blades and guides as computed are laid out in Fig. 52, which shows the arrangements and methods of securing them. Since the peripheral speed is high and the impulse from the jets is

energetic, the blades and guides must be made of steel or strong bronze. The details are similar to those for the Curtis turbine.

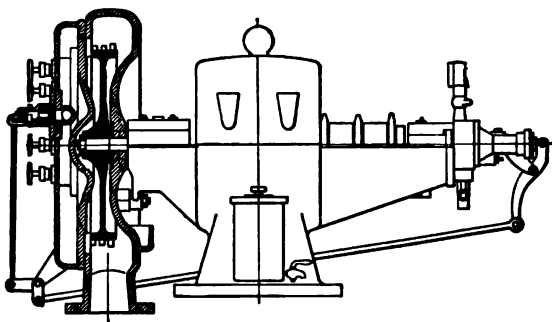


FIG. 53.

**General Arrangement.** — The general arrangement of a turbine of the type designed is shown by Fig. 53, which shows a single stage turbo-generator built by the Allgemeine Elektrizitäts-

**Gesellschaft.** The electric generator is set between bearings and the turbine is overhung. The nozzles are set in the out-board cover of the turbine casing and are controlled by a governor similar to that explained on page 231. After passing through three sets of blades and the intermediate guides the steam is exhausted downwards; the blades and guides are not well represented in the figure.

**Modifications.** — The computations and constructions for the design of a velocity compound turbine have been carried through systematically in the usual manner with all the ordinary refinements. Leaving at one side the discussion of the advisability of designing a turbine with the very high velocities computed, there are certain practical considerations that should receive attention. In the first place the nice adjustment of the peripheral width of the blade edges and the spaces occupied by the walls between adjacent nozzles will probably receive scant attention from a practical designer. The thickness of the blade edges is commonly made 0.02 of an inch; the exit length of a blade may be computed much as in the design; the entrance length will be made as great as the radial dimension of the nozzle, or as great as the exit length of the preceding guide, as the case may be, and guides will be treated in the same way.

But many designers increase the lengths of blades and guides beyond those calculated by our methods (or their equivalents), so that there may be no danger of choking the flow of steam, especially for the blades receiving steam from the nozzles. And further rapid changes of length, as for the guides of Fig. 52, are avoided by arbitrarily increasing dimensions at discretion.

**Repeated Flow Turbine.** — A simple and compact form of turbine with velocity compounding is obtained by causing the steam to flow back and forth through the same blades, as shown by Fig. 54. The steam is expanded from the initial pressure to the back pressure in a nozzle *N*, and passes through the blades of the wheel into a curved passage or guide *G*, which leads the steam back to the same set of blades; after passing through the blades again the steam is led by the passage or guide *G* to the

third passage through the blades, and the passage or guide *G* then leads it to the fourth passage through the blades, and so to the exhaust. The four passages through the blades of the wheel correspond to the passage through four sets of blades like those of Fig. 44. As the relative velocity through the blades decreases, the widths of the passages increase and more blades are affected;

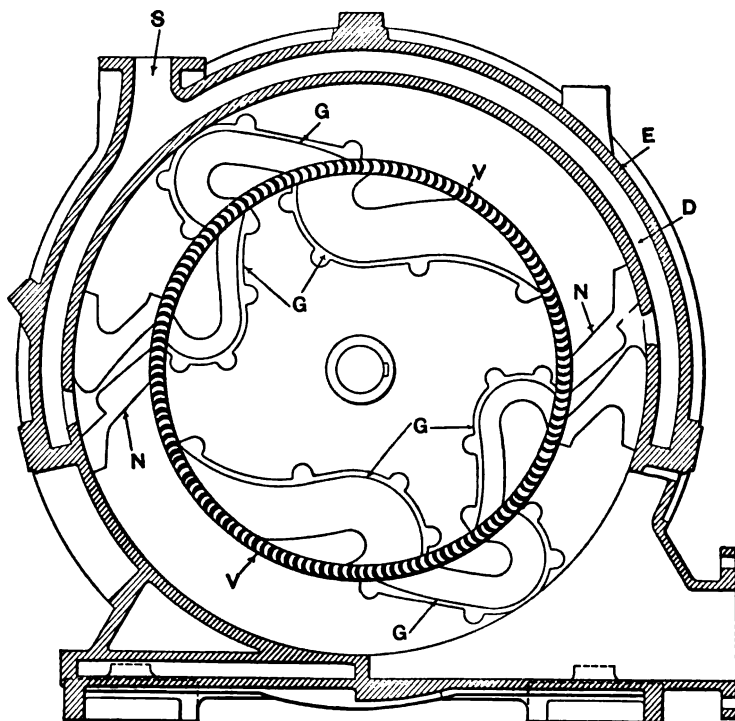


FIG. 54.

it will be seen later that the angles of the guides diminish for the successive passages, which further increases the number of blades in action; and finally the effect of friction is to dry and expand the steam, for which allowance must be made. In Fig. 54 there are two nozzles and the dimensions are so chosen that the blades are, as nearly as may be, in continuous action. This figure, which illustrates the Electra turbine, is chosen because it shows

clearly the arrangement of guides and blades; since the steam passes in and out through the blades it is affected by centrifugal force, which alternately decreases and increases the exit velocity from the blades. But since the width of the blades is small compared with the radius of the wheel, the effect of centrifugal force is not important, and can either be neglected or taken care of by increasing or decreasing the factors for blade friction.

In some cases the blades are set radially on the edge of the wheel, and the steam passes through them from right to left and back again, as in Fig. 55, which represents the sections of such

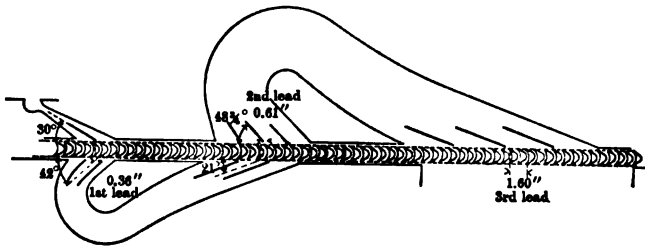


FIG. 55.

radial blades by a cylindrical surface at the pitch circle, developed so as to bring them into the plane of the paper. Such an arrangement avoids the influence of centrifugal force and allows of a direct and simple graphical construction for the angles and velocities.

**Angles and Velocities.** — The diagram of velocities and angles can be constructed as shown in Fig. 56, which provides for three passages of the steam through the blades. The controlling feature of the construction is the fact that the blade angle remains constant, since the steam passes back and forth through the same blades. The direct construction of the diagram will be described first; afterward an inverse construction, which will be found more convenient in practice, will readily be understood. Beginning with the jet velocity  $V_1$  and the nozzle angle  $\alpha$ , the triangle  $V_1, V, V_2$ , or  $oab$ , is drawn to determine the relative

velocity  $V_2$  and the blade angle  $\beta$ , which latter remains constant as seen by inspection of the diagram. Assuming a coefficient for friction  $\gamma_b$ , the relative velocity at exit  $V_3$  is made equal to

$$\sqrt{1 - \gamma_b} V_2,$$

and then the triangle  $V_3, V, V_4$ , or  $ocd$ , is drawn to determine the absolute velocity  $V_4$  with which the steam enters the first guide or passage.

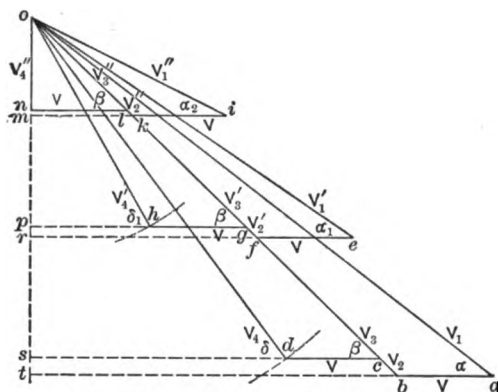


FIG. 56.

The absolute velocity  $V_4$  is to be affected by the coefficient for friction  $\gamma_o$ , so that the absolute exit velocity of the steam leaving the passage is

$$\sqrt{1 - \gamma_o} V_4 = V_1'.$$

Since the passages are relatively long a comparatively large value may be assigned to  $\gamma_o$ .

Having the velocity  $V_1'$  and the peripheral velocity  $V$ , the triangle  $V_1', V, V_2'$ , or  $oef$  is to be constructed by trial so that  $of$  shall coincide with  $ob$ ; since the direct construction is really a descriptive process, the details of such construction need not be considered. This construction determines  $\alpha_1$  and  $V_2'$ ; having  $V_2'$ ,  $V_3'$  can be found by allowing for friction and afterward  $V_4'$  may be determined by drawing the triangle  $V_3', V, V_4'$ . A

repetition of the process will determine the remainder of the velocities and the angle  $\alpha_2$ . The construction would require to be repeated if necessary to make  $V_4''$  vertical or nearly so, but such a process would be so tedious that the inverse construction is to be preferred.

In anticipation of a design for turbines of this type, it is desirable to make tables of velocities and angles (similar to those given for velocity compounding) with several sets of blades; the construction of such tables can best be carried on by an inverse construction beginning with the last passage through the blades. For this purpose a convenient length may be taken for  $V$ , the peripheral velocity which may be called unity; all the other velocities can be determined in terms of this unit and afterwards the ratios of velocities to the initial velocity can be found.

A convenient value for  $\beta$ , the blade angle, may be assumed (from  $30^\circ$  to  $45^\circ$ ). The triangle  $oln$  may be drawn with this value of  $\beta$  and with the assumed unit value for  $V$ , thus determining the relative velocity  $V_3''$  at exit from the blades; the relative entrance velocity is

$$V_2'' = V_3'' \div \sqrt{1 - \gamma_b},$$

where  $\gamma_b$  is the coefficient for blade friction. With  $V$ , the unit velocity, construct the triangle  $oki$ , thus determining the absolute velocity  $V_1''$  with which the steam leaves the last guides or passage, and the angle  $\alpha_2$  of those guides.

The absolute velocity  $V_4'$  with which the steam enters the last guides or passage is

$$V_4' = V_1'' \div \sqrt{1 - \gamma_g},$$

where  $\gamma_g$  is the coefficient of friction for the passage; as already indicated this coefficient is relatively large. To construct the triangle  $gho$ , draw an arc with the radius  $oh$ , equal to  $V_4'$ , from the center  $o$ , and find by trial the position of the line  $hg$ , so that it shall be horizontal and reach from the arc to the line  $og$ . Having the triangle  $ohg$ , the remainder of the construction for the second passage through the blades is obvious. The angle

of entrance to the last passage is  $\delta_1$  and the angle of exit from the first guides is  $\alpha_1$ . A repetition of the construction with an arc drawn from  $o$  through  $d$  will complete the diagram.

Since the inverse construction just stated has the disadvantage of proceeding from small velocities to large, and of a somewhat indefinite location of the lines  $hg$  and  $cd$ , it is recommended that the angles and velocities be computed. Such a computation beginning with the triangle  $V_4''VV_3''$  is direct trigonometrical work which can but be illustrated by an example.

Assuming a blade angle of  $35^\circ$  and the friction factors  $y_b = 0.1$  and  $y_g = 0.2$  for the blades and guides, we may proceed to find the ratios of the velocities, the guide angles, and the efficiencies as follows. The peripheral velocity  $V$  will be taken as unity and the friction factors will be

$$\sqrt{1 - y_b} = \sqrt{1 - 0.1} = 0.9486; \quad \sqrt{1 - y_g} = \sqrt{1 - 0.2} = 0.8945.$$

In Fig. 56 the velocities for the last passage of steam through the blades are

$$V_4'' = V \tan 35^\circ = 0.7002 V; \quad V_3'' = V \sec 35^\circ = 1.2208 V; \\ V_2'' = V_3'' \div \sqrt{1 - y_b} = 1.2208 V \div 0.9486 = 1.287 V.$$

In the triangle  $okm$

$$mk = V \div \sqrt{1 - y_b} = V \div 0.9486 = 1.054 V; \\ om = V_4'' \div \sqrt{1 - y_b} = 0.7002 V \div 0.9486 = 0.7380 V.$$

In the triangle  $oim$

$$mi = mk + V = 1.054 V + V = 2.054 V; \\ \tan \alpha_1 = om \div mi = 0.7380 \div 2.054; \quad \therefore \alpha = 19^\circ-46'; \\ V_1'' = mi \div \cos \alpha = 2.054 V \div \cos 19^\circ-46' = 2.182 V.$$

Coming to the triangle  $ohg$  we have

$$V_4' = V_1'' \div \sqrt{1 - y_g} = 2.182 V \div 0.8945 = 2.440 V; \\ V_4' : V :: \sin \beta : \sin goh = V \sin 19^\circ-46' \div 2.440 V; \\ \therefore goh = 13^\circ-36'; \\ \delta_1 = goh + \beta = 13^\circ-36' + 35^\circ = 48^\circ-46'; \\ ph = V_4' \cos \delta_1 = 2.440 V \cos 48^\circ-46' = 1.614 V;$$

$$\begin{aligned} pg &= 1.614 V + V = 2.614 V; \\ V_3' &= pg \div \cos \beta = 2.614 \div \cos 35^\circ = 3.191; \\ V_2' &= V_3' \div \sqrt{1 - y_b} = 3.191 V \div 0.9486 = 3.364 V. \end{aligned}$$

In the triangle *orf*

$$\begin{aligned} rf &= pg \div \sqrt{1 - y_b} = 2.614 V \div 0.9486 = 2.756 V; \\ or &= V_2' \sin \beta = 3.364 V \sin 35^\circ = 1.929 V; \end{aligned}$$

and in the triangle *ore*

$$\begin{aligned} re &= rf + V = 2.756 V + V = 3.756 V; \\ \tan \alpha_1 &= or \div re = 1.929 \div 3.756; \therefore \alpha_2 \div 27^\circ - 8'; \\ V_1' &= re \div \cos \alpha_1 = 3.756 V \div \cos 27^\circ - 8' = 4.218 V; \\ V_4 &= V_1' \div \sqrt{1 - y_o} = 4.218 V \div 0.8945 = 4.715 V. \end{aligned}$$

Now we have in the triangle *ocd*

$$\begin{aligned} V_4 : V_1 :: \sin \beta : \sin cod &= V \sin 35^\circ \div 4.715 V; \therefore cod = 7^\circ; \\ \delta &= \beta + cod = 35^\circ + 7^\circ = 42^\circ; \\ sd &= V_4 \cos \delta = 4.715 V \cos 42^\circ = 3.500 V; \\ sc &= sd + V = 3.500 V + V = 4.500 V; \\ V_3 &= sc \div \cos 35^\circ = 4.500 V \div \cos 35^\circ = 5.493 V; \\ V_2 &= V_3 \div \sqrt{1 - y_b} = 5.493 V \div 0.9486 = 5.790 V. \end{aligned}$$

In the triangle *otb*

$$\begin{aligned} tb &= sc \div \sqrt{1 - y_b} = 4.500 V \div 0.9486 = 4.744 V; \\ ot &= V_2 \sin \beta = 5.790 V \sin 35^\circ = 3.321 V; \end{aligned}$$

and in the triangle *oat*

$$\begin{aligned} ta &= tb + V = 4.744 V + V = 5.744 V; \\ \tan \alpha &= ot \div ta = 3.321 \div 5.744; \therefore \alpha = 30^\circ - 3'; \\ V_1 &= ta \div \cos \alpha = 5.744 \div \cos 30^\circ - 3' = 6.636 V. \end{aligned}$$

To get the efficiency we must first find the total retardation as follows:

$$\begin{array}{rcl} ta + sc &= 5.744 V + 3.500 V &= 9.244 V \\ re + ph &= 3.756 V + 1.614 V &= 5.370 V \\ mi + o &= 2.054 V + 0 &= 2.054 V \\ \hline \text{Total} && 16.668 V \end{array}$$

The retardation  $sc$  and  $ph$  have properly a negative sign, so that on subtraction they give the arithmetical sums as indicated. The total is now to be multiplied by the peripheral speed  $V$ , and divided by  $\frac{1}{2} V_1^2$ , so that the efficiency is

$$16.668 \times 2 \div 6.636^2 = 0.745.$$

It is convenient for design to have the various velocities expressed in the form of the ratio to the velocity of the jet  $V_1$ ; so the several values found are divided by 6.636  $V$  and are here assembled.

#### RATIOS OF VELOCITIES TO JET VELOCITY

$V$ , 0.151	$V_1'$ , 0.636	$V_1''$ , 0.329
$V_2$ , 0.873	$V_2'$ , 0.507	$V_2''$ , 0.194
$V_3$ , 0.828	$V_3'$ , 0.481	$V_3''$ , 0.184
$V_4$ , 0.711	$V_4'$ , 0.368	$V_4''$ , 0.106
$\alpha$ , $30^\circ$	$\alpha_1$ , $27^\circ$	$\alpha_2$ , $19^\circ\frac{1}{2}$
$\delta$ , $42^\circ$	$\delta_1$ , $48^\circ\frac{1}{2}$	$\delta_2$ , $90^\circ$
Efficiency 0.75.		

**Design of a Repeated-Flow Turbine.** — Let it be required to design a triple-flow turbine with the following conditions:

Brake horse-power . . . . .	300
Pressure, gauge, pounds . . . . .	138.2
Vacuum, inches . . . . .	26
Angle of blades $\beta$ . . . . .	$35^\circ$
Friction factors, for nozzles $y_n$ . . . . .	0.10
for blades $y_b$ . . . . .	0.10
for passages $y_p$ . . . . .	0.20
Heat losses due to rotation, etc. . . . .	0.85
Mechanical efficiency . . . . .	0.90

This design, like the previous one, is worked out as an illustration and the results have not the support of comparison with practice. The conditions, so far as may be, are those of the preceding design, so that some of the results of computations for that design may be stated at once for this.

The absolute pressure at entrance to the nozzle, at the throat, and at exit are 152.9, 88.7, and 2 pounds, the corresponding temperatures are  $360^\circ$ ,  $319^\circ$ , and  $126^\circ$ ; the entropy at entrance is 1.56.

The steam velocities for the nozzle are: throat 1483, and at exit 3566 feet per second; the latter being the velocity  $V_1$  of

entrance to the blades at the first passage. The efficiency for the Rankine cycle is 0.258. The blade efficiency for the conditions of this design as computed is 0.75. The efficiency of the turbine not allowing for gland leakage or radiation is

$$\begin{array}{cccccc} \text{Nozzle} & \text{Thermal} & \text{Blade} & \text{Disk} & \text{Mech.} & \\ 0.9 \times 0.258 \times 0.75 \times 0.85 \times 0.9 = 0.133. \end{array}$$

The thermal units per horse-power per minute are

$$42.42 \div 0.133 = 319 \text{ B.T.U.},$$

and since the heat required to vaporize a pound of steam is

$$x_1 r_1 + q_1 - q_s = 1187.1 - 94 = 1093.1 \text{ B.T.U.},$$

there will be

$$319 \times 60 \div 1093.1 = 17.5$$

pounds of steam per brake horse-power per hour.

The steam per second will be

$$\frac{17.5 \times 300}{60 \times 60} = 1.46 \text{ pounds.}$$

The specific volumes at the throat and at the exit, as given on page 134, are 4.73 and 142.8 cubic feet. Consequently, the total throat area and exit area will be

$$a_t = \frac{144 \times 4.73 \times 1.46}{1483} = 0.67 \text{ square inches;}$$

$$a_e = \frac{144 \times 142.8 \times 1.46}{3566} = 8.42 \text{ square inches.}$$

If we take two nozzles they will have for the areas of each nozzle in square inches

Throat 0.335;

Exit 4.21.

The throat may have a circular section with the diameter 0.653 of an inch. The exit may have the section approximately square; if the radial dimension is two inches, the net width will be 2.11 inches.

The computations of relative velocities and angles on page 150 are for the conditions of this design and the results therefore apply directly. In the following table the several velocities are found by multiplying the jet velocity by the factors there given.

	Velocity.	Heat equivalent.	Diff.	Heat contents	Volumes.	Angles.	Sines.
Nozzle $V_1$ . . . . .	3566	254.0	....	933.1	142.8	30°	0.5000
Peripheral velocity $V$ . . . . .	538	....	....	....	....	....	....
Wheel, entrance $V_2$ . . . . .	3114	193.8	....	933.1	142.8	35°	0.5736
exit $V_3$ . . . . .	2954	174.4	19.4	952.5	146.1	35°	....
1st guide, entrance $V_4$ . . . . .	2536	128.3	....	952.5	146.1	42°	0.6691
exit $V_1'$ . . . . .	2268	102.8	25.5	978.0	150.5	27°	0.4540
Wheel, entrance $V_2'$ . . . . .	1808	65.3	....	978.0	150.5	35°	0.5736
exit $V_3'$ . . . . .	1716	58.9	6.4	984.4	151.6	35°	....
2d guide, entrance $V_4'$ . . . . .	1313	34.5	....	984.4	151.6	48° $\frac{1}{2}$	0.7518
exit $V_1''$ . . . . .	1173	27.5	7.0	991.4	152.8	19° $\frac{1}{2}$	0.3379
Wheel, entrance $V_2''$ . . . . .	692	9.6	....	991.4	152.8	35°	0.5736
exit $V_3''$ . . . . .	656	8.6	1.0	992.4	153.0	35°	....
Final absolute $V_4''$ . . . . .	378	....	....	992.4	153.0	90°	1.000

Having the velocities, the heat equivalents are computed by the equation

$$\text{Heat equivalent} = V^2 \div 2g \times 778.$$

The initial heat contents (see pages 133 and 140) was 1187.1 B.T.U., of which 254.0 were changed into kinetic energy at exit from the nozzle, leaving

$$1187.1 - 254.0 = 933.1 \text{ B.T.U.}$$

per pound of steam in the jet. To this was added 19.4 B.T.U. by friction during the first passage of the wheel, and 25.5 B.T.U. in traversing the first guide or passage, and so forth for the other passages through the wheel and guides.

The specific volumes are found by interpolation in the entropy table at 126°.

The net width of the nozzles has been found to be 2.11 inches; the effective areas, and consequently the net widths of the entrances and exits to the guides or passages, will be inversely proportional to the absolute velocities at those places, and

directly proportional to the specific volumes. Consequently, we shall have for the net widths first guides or passages

$$\text{Entrance} \quad 2.11 \times \frac{3566}{2536} \times \frac{146.1}{142.8} = 3.035;$$

$$\text{Exit} \quad 2.11 \times \frac{3566}{2268} \times \frac{150.5}{142.8} = 3.495;$$

Second guides or passages

$$\text{Entrance} \quad 2.11 \times \frac{3566}{1313} \times \frac{151.6}{142.8} = 6.08;$$

$$\text{Exit} \quad 2.11 \times \frac{3566}{1173} \times \frac{152.8}{142.8} = 6.86.$$

The net peripheral space on the pitch circle of the wheel will be found by dividing by the sines of the angles as follows:

$$\text{Nozzle, exit} \quad 2.11 \div 0.5000 = 4.22 \text{ inches};$$

First guides or passages

$$\text{Entrance} \quad 3.035 \div 0.6691 = 4.54 \text{ inches};$$

$$\text{Exit} \quad 3.495 \div 0.4540 = 7.70 \text{ inches};$$

Second guides or passages

$$\text{Entrance} \quad 6.08 \div 0.7518 = 8.09 \text{ inches};$$

$$\text{Exit} \quad 6.86 \div 0.3379 = 20.30 \text{ inches}.$$

It is now necessary to determine the dimensions of the blade by aid of a drawing, like Fig. 57, which is drawn with an axial length of 1.5 inches, a pitch of 0.9 of an inch, and with the blade edge 0.03 of an inch, measured on the periphery. This gives for the blank peripheral space  $3\frac{1}{2}$  per cent. As is shown on page 95, the same proportion of blank space should be assigned to the ends of the guides, if the radial height of blades and guides is to be the same. Such an arrangement will allow of division plates in nozzles and passages. The gross peripheral spaces for the nozzles and guides or passages will be found by dividing by

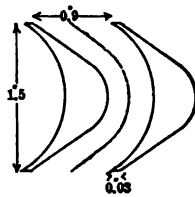


FIG. 57.

$$1 - 0.033 = 0.967.$$

Gross peripheral spaces, nozzles and guides, inches:

Nozzle, exit  $4.22 \div 0.967 = 4.36$  inches:

First guides or passages

Entrance  $4.54 \div 0.967 = 4.70$  inches;

Exit  $7.70 \div 0.967 = 7.96$  inches;

Second guides or passages

Entrance  $8.09 \div 0.967 = 8.37$  inches;

Exit  $20.30 \div 0.967 = 21.00$  inches.

If the wheel discharges into a closed exhaust space, its peripheral dimension will be found in like manner, from the absolute velocity  $V_4''$ . The net width, which is also the net peripheral space, since the angle is  $90^\circ$ , is

$$2.11 \times \frac{3566}{378} \times \frac{153.0}{142.8} = 21.3 \text{ inches,}$$

and the gross peripheral space is

$$21.3 \div 0.967 = 22.1 \text{ inches.}$$

The blank space along the pitch periphery has been determined to be 0.0333 of the gross peripheral dimensions of nozzles and passages. The total thickness of plates can be found by multiplying such spaces by the sine of the proper angles, as follows:

Nozzle	$4.36 \times 0.0333 \times 0.5000 = 0.073''$
First guide, entrance	$4.70 \times 0.0333 \times 0.6691 = 0.105$
exit	$7.96 \times 0.0333 \times 0.4540 = 0.120$
Second guide, entrance	$8.37 \times 0.0333 \times 0.7518 = 0.210$
exit	$21.00 \times 0.0333 \times 0.3379 = 0.237$
Exhaust	$22.10 \times 0.0333 \times 1.0000 = 0.736$

An exact adjustment in this place would be troublesome, but a fair approximation could be had by dividing the exit of the nozzle by one plate of No. 15 U.S. gauge, and using two plates gauge 17 in the first passage and three plates gauge 15 in the second passage; it probably would not be desirable to put plates in the exhaust passage.

In locating the guides or passages allowance must be made for the lead as explained on page 103. The mean path followed by the steam, as shown by the dotted line in Fig. 57, is about two inches. This is traversed at the mean of the steam speeds at entrance and exit to the blades, and the wheel meanwhile moves with the peripheral speed; the lead is therefore

$$2 \times \frac{V}{\frac{1}{2}(V_2 + V_3)} = 2 \times 538 \div 3034 = 0.36 \text{ inches.}$$

The leads for the second and third passages of steam through the wheel are

$$2 \times 538 \div 1762 = 0.61 \text{ inches; } \quad 2 \times 538 \div 674 = 1.60 \text{ inches.}$$

The developed cylindrical section through the nozzle and passages is diagrammatically laid out in Fig. 55, page 147. The nozzle has an angle of  $30^\circ$  and the center line is drawn to intersect the dotted line, which indicates the edges of the blades; a vertical line is drawn to the dotted line, representing the opposite edges of the blades; the lead, 0.36 of an inch for the first passage of the steam, is laid off. From the point so located the center line for the first guide or passage is laid off, making the angle of  $42^\circ$ . The passage is drawn with smooth curves and gradually increasing width as required by the design. The exit from the guides has the angle of  $27^\circ$ , and the center line is extended to the lower edge of the blades, and from the intersection a vertical line is drawn to the line of the upper edges of the blades; here the lead 0.61 of an inch for the second passage through the blades is laid off, and the center line is drawn, making the angle of  $48\frac{1}{4}^\circ$ . At this place the design exhibits a discrepancy, because the peripheral space for the entrance to the second passage is less than that for the exit from the first passage; to avoid choking, the passage at entrance is arbitrarily made as wide as the preceding exit, but is then contracted to its proper width. The lead 1.6 of an inch is laid off to find the center of the exhaust passage. The nozzle is indicated to have one dividing plate, the first passage has two and the second passage has three.

The overall length of the construction is about 66 inches; two nozzles with their passages will, therefore, require something more than ten feet. If the turbine be given 3000 revolutions per minute the diameter will be

$$d = \frac{60 \times 538}{3000 \pi} = 3.42 \text{ feet} = 3 \text{ feet } 5 \text{ inches.}$$

We may now proceed to assign the number of blades and to correct the pitch if necessary. The periphery of the wheel at the pitch surface will be

$$\pi 3.42 = 10.75 \text{ feet or } 129 \text{ inches;}$$

if 144 blades are assigned to the wheel, the pitch will be 0.896, which is slightly less than the 0.9 provisionally taken.

The construction of Fig. 55 is characterized by long passages and extensive spaces between them. The guide plates are drawn only at the entrances and exits of the passages; they should have sharp edges in the passages, and the passages should be somewhat narrower between them.

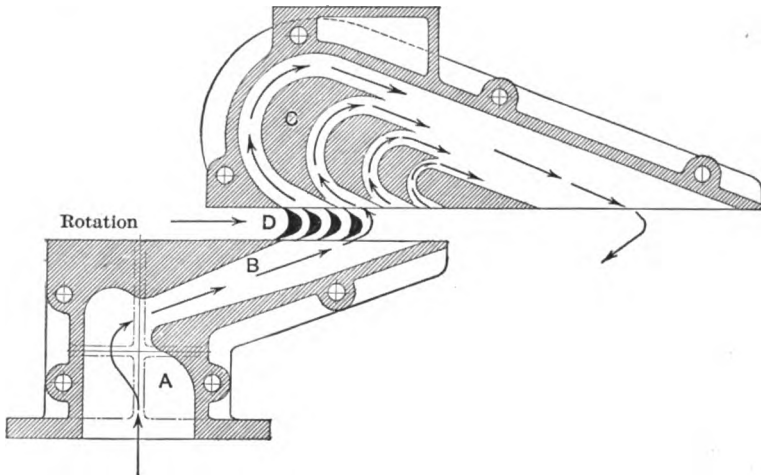


FIG. 58.

A method actually used for reducing the spaces between the passages for a repeated-flow turbine is shown by Fig. 58.

The figure represents the nozzles and passages for a turbine which passes the steam twice through the blades and exhausts against a back pressure of ten pounds. The makers of this turbine give importance to reducing the clearance between the rotor and the nozzle-and-passages chamber to the least possible amount. The clearance proper is made 0.015 of an inch, and is further reduced by packing strips forced out by springs; these strips are machined to touch the rotor, but wear away to give a clearance of 0.005 of an inch, being held by shoulders from moving closer.

**Double-Motion Turbine.** — The intermediate guides of a velocity-compound turbine can be omitted, if the successive wheels are driven in opposite directions. This type appears to be adapted only for small powers like the Seger turbine, shown by Fig. 59; the use of more than two wheels would involve mechanical complications.

This type of turbine can be designed to give either equal revolutions for the wheels or equal powers; the Seger turbine has nearly equal powers for the wheels. The arrangement with equal revolutions, which conforms more directly with previous discussions, is shown by Fig. 60; here the velocity of the first wheel  $V$  is made one-fourth the initial velocity of whirl, that is, equal to one-fourth of the line  $ab$ . The triangle  $V_1VV_2$  determines the relative velocity with which the steam enters the blades of the first wheel. Assuming that the blade angles are equal, and neglecting steam friction, the relative exit velocity is made equal to  $V_2$  and laid off at  $V_3$ . The triangle  $V_3VV_4$  determines the absolute exit velocity from the first wheel, and this is the absolute velocity  $V_1'$  with which the steam strikes the second wheel. The second wheel moves in the contrary direction, that is, toward the left; the triangle  $V_1'VV_2'$  determines the relative velocity  $V_2'$  with which the steam enters the blades of the second wheel,  $V$  being the same as for the first wheel, but directed toward the left. The absolute exit velocity of steam leaving the second wheel  $V_4'$ , determined by the triangle  $V_3'VV_4'$ , is vertical, so that there is no residual velocity of whirl. The

outlines of the blade sections are drawn by the usual methods. This arrangement is interesting, because it has been used for driving the propellers of automobile torpedoes, which are mounted on one shaft and revolve in contrary directions, so that the torque

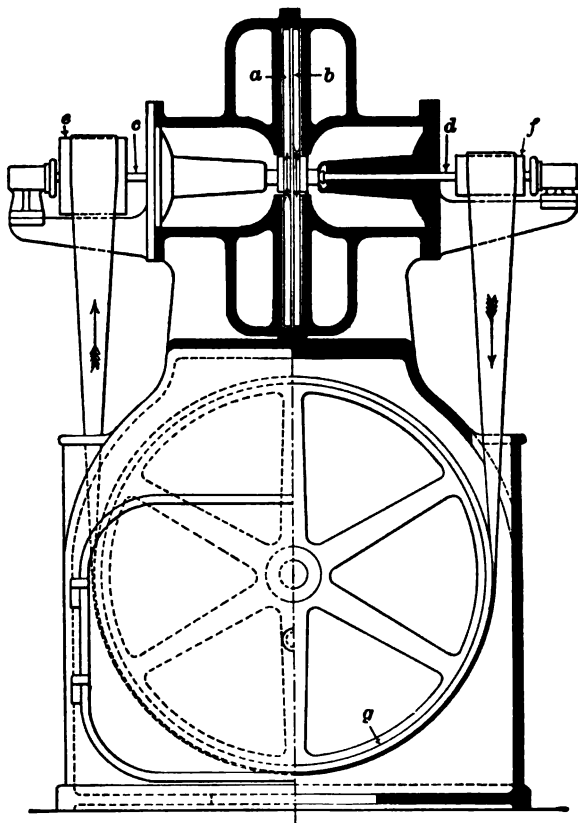


FIG. 59.

may not affect the stability of the torpedo; one propeller is fixed directly to the shaft and the other to a sleeve, with toothed gearing to ensure that the revolutions shall be equal and contrary. The turbine wheels are conveniently fixed, one to the shaft and the other to the sleeve, and are driven by compressed air from a reservoir.

The work of the second wheel is only one-third of that of the first wheel, as can be seen from the following discussion. The initial velocity of whirl is  $ab$ , equal to  $4V$ ; the velocity of whirl of the steam at exit from that wheel is  $dc$ , equal to  $-2V$ . The retardation is therefore

$$ab - (-dc) = 4V - (-2V) = 6V.$$

The velocity of whirl of the steam entering the second wheel is  $dc$  toward the left, and the velocity of whirl at exit is zero. Consequently the retardation in the second wheel is

$$dc - 0 = 2V - 0 = 2V.$$

But since the peripheral velocity is  $V$  for each wheel, the works are proportional to the retardations, which have been shown to be in the ratio

$$6V : 2V = 3.$$

In Fig. 61, which is drawn without taking account of steam friction and assuming the blade angles are equal for both wheels, the velocity  $V$  of the first wheel

is made equal to  $\frac{1}{n}$  part of  $ab$ , the initial velocity of whirl; conversely  $ab$  is  $n$  times  $V$ . The exit velocity of whirl  $dc$  is

numerically equal to  $(n - 2)V$  and the retardation is

$$nV - [-(n - 2)V] = (2n - 2)V, \quad \dots (1)$$

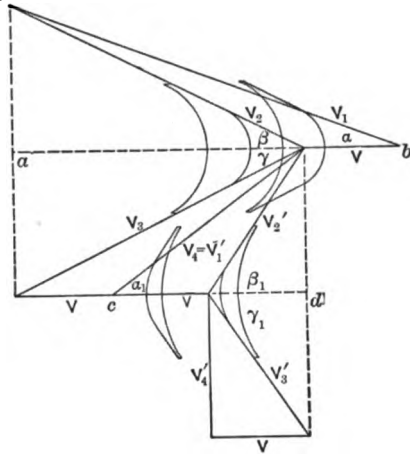


FIG. 60.

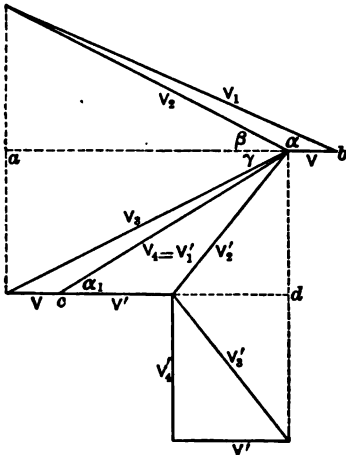


FIG. 61.

so that the work of the first wheel is proportional to

$$(2n - 2)V^2 \quad \dots \quad (2)$$

Since the velocity of whirl at entrance to the second wheel is

$$dc = (n - 2)V \quad \dots \quad (3)$$

and at exit is zero, the retardation is represented by (3). The velocity of the wheel is half of  $dc$ , and consequently the work of the second wheel is proportional to

$$\frac{1}{2} (n - 2)V (n - 2)V = \frac{1}{2} (n - 2)^2 V^2 \quad \dots \quad (4)$$

This expression must be equal to (2), if the works are to be equal, consequently

$$\begin{aligned} 2n - 2 &= \frac{1}{2} (n - 2)^2; \\ \therefore 4n - 4 &= n^2 - 4n + 4; \\ n &= 4 + \sqrt{16 - 8} = 6.828. \end{aligned}$$

The ratio of the velocities is

$$V : V' = 1 : \frac{1}{2} (n - 2) = 1 : 2.414.$$

In Fig. 59 the wheels are overhung on the shafts which carry the pulleys  $ec$  and  $df$ ; the steam passes first through the wheel  $a$  and then through the wheel  $b$ ; the pulley for  $b$  has half the diameter of the other and it therefore makes twice the number of revolutions. A figure like Fig. 61 can be drawn with  $V$  equal to one-sixth of  $ab$ , and  $V'$  equal to one-third of that dimension; this arrangement distributes the work in the ratio of 10 to 8.

The relations given for equal velocity or equal works will be slightly affected if allowance is made for friction of the steam; if the exit angles of the wheels are less than the entrance angles the relations will be affected in a marked manner, and there will be appreciable end thrust.

In any case, the peripheral velocity of the wheels will be high and especially for the second wheel of the combination to give equal works; and there is likely to be trouble from vibration, especially as flexible shafts do not appear applicable. Since

the power will be small a few nozzles will suffice and they will be symmetrically disposed around the first wheel. The nozzles can be made like Fig. 3, page 20, reamed to a straight cone and continued by a cylindrical shell. The blades will be slightly longer than the diameter at exit, and those of the second wheel a little longer than those of the first. If the exit angles are reduced then the blade lengths may require investigation; otherwise refinement appears unnecessary.

## CHAPTER VII

### PRESSURE AND VELOCITY COMPOUNDING

GREAT range and flexibility of design and application of steam turbines for all purposes can be had by a combination of pressure compounding and velocity compounding. The exponent of this type is the Curtis turbine, represented by Fig. 62 and by Fig. 63. Large, stationary turbines are arranged with the axis vertical, as shown by Fig. 62, which has the condenser in the base directly under the turbine, while the electric generator is overhead. Smaller powers have horizontal axes like Fig. 63, which represents the 500 K.W. turbine with three bearings for the shaft, one being beyond the generator. Small turbines with only two pressure stages commonly have but two shaft bearings, the turbine being overhung like the velocity-compound turbine of Fig. 53, page 144. Marine turbines like Fig. 72, facing page 191, have a large number of stages in order to get low peripheral velocity.

For stationary purposes the arrangements vary from a type with two pressure stages each having three rows of moving blades, to a type having six pressure stages each having two rows of blades. The marine type has four rows of blades in the first pressure stage, and three rows in other stages; the low-pressure stages are on a drum and may have two rows of blades per stage, or in some cases only one, that part of the turbine reverting to the Rateau type.

**Nozzles, Blades, and Guides.** — The general arrangement of nozzles, blades, and guides for a pressure and velocity compound turbine is similar to that discussed on page 100 *et seq.* for velocity compound turbines; since there are a number of pressure stages the drop of pressure per stage is less, and the nozzles and guides can more easily be satisfactorily arranged.

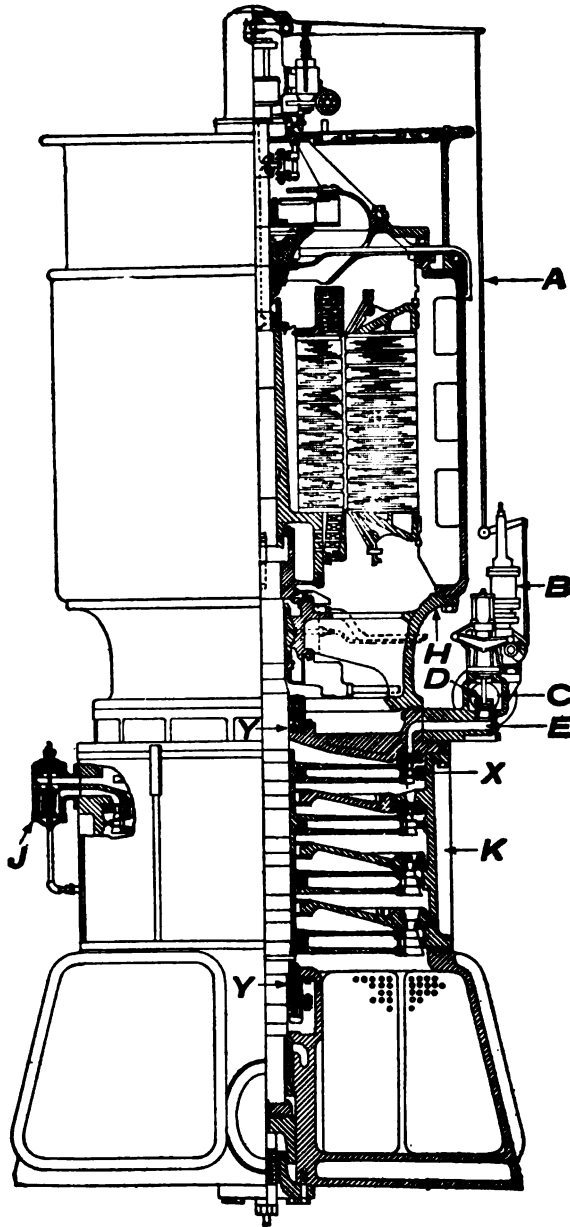


FIG. 62.

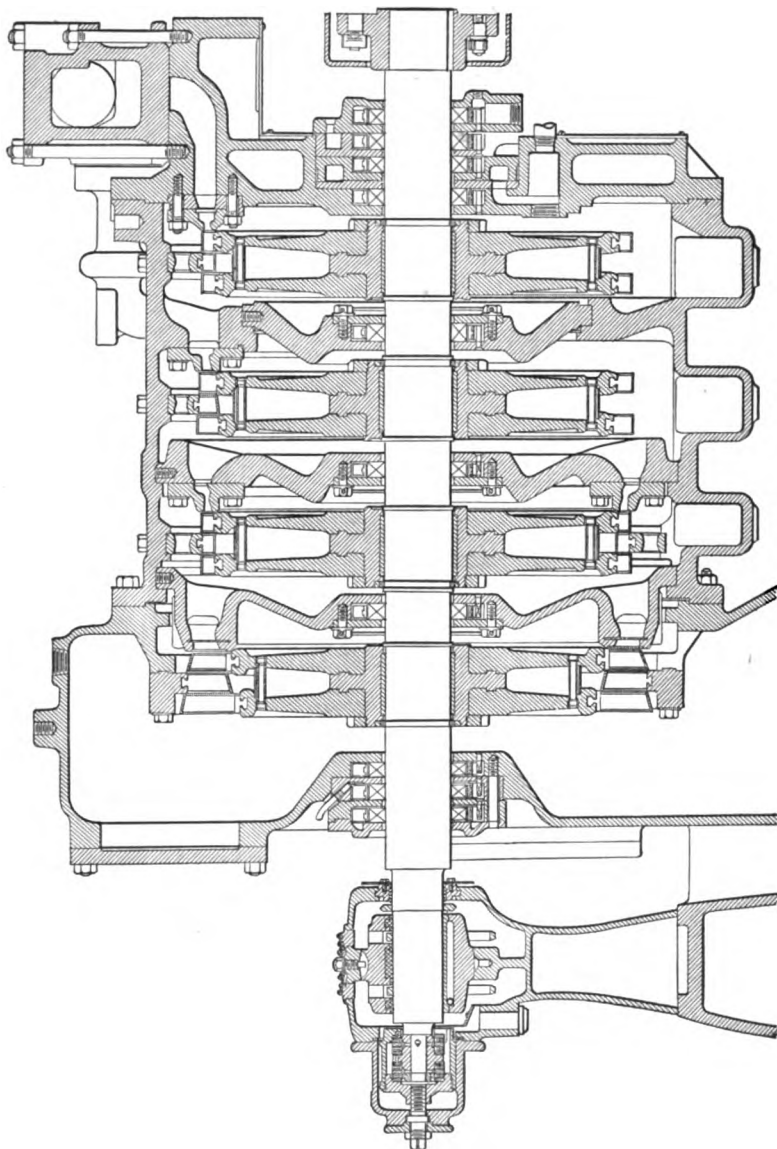


FIG. 63. CURTIS FOUR-STAGE TURBINE.

Curtis turbines, as built by the General Electric Company, have until recently had expanding nozzles filed from a solid casting, much as shown by Fig. 49, page 136, and accompanying description, except that the throat was square as well as the exit. More recently such expanding nozzles are reamed with a straight, conical seamer. The two-stage turbine expands steam from boiler-pressure nearly down to atmospheric-pressure in the first set of nozzles, the nozzles of that set being equivalent to those used for a non-condensing de Laval turbine. The second set of nozzles farther expands the steam down to the vacuum in the exhaust passage. Both sets of nozzles have considerable expansion from the throat to the exit but can be made conveniently by filing or reaming. Turbines with four or more stages have comparatively little expansion in any of the nozzles; the nozzles for the first two stages are commonly filed or reamed; the successive nozzles may readily be made with plates, cast or machined into the supporting castings, as illustrated by the dotted lines of Fig. 26, page 89. The marine turbine of the Curtis type is likely to have twelve or more pressure stages. Of these the first stage has expanding nozzles, which drop the pressure from boiler-pressure to about 75 pounds absolute; the other stages have little, if any, expansion in the nozzles, which are commonly made with uniform area from the throat to the exit.



FIG. 65.

There are three types of construction for moving blades and stationary guides of turbines. When the peripheral velocity and steam impulses are comparatively small, the blades and guides are made of extruded metal and dovetailed into a groove as shown by Figs. 64 and 65; the first figure shows a segment of blades which is bolted to the edge of a revolving wheel, the shoulder taking the centrifugal force; the second figure gives the plan and elevation of one blade, which has a tip at the end

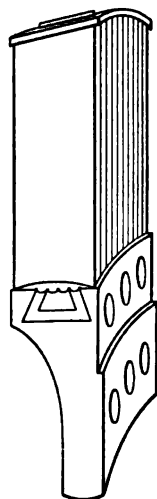


FIG. 64.

protruding through a shrouding plate and is riveted over as shown by Fig. 64. The blades are separated by filling pieces, also dovetailed into the groove and shaped to conform to the backs and faces of the blades. When the peripheral speed is higher, the dovetailing of the root of the blade is modified to show a neck and shoulder, as indicated by Fig. 53, page 144, but the blades are still made of extruded metal and separated by filling pieces. When the peripheral speed becomes as high as 500 feet per second, the blades are drop forged in one piece with the filler attached, from manganese bronze as shown by Fig. 66;

this type can run at a peripheral speed of 600 feet per second.

For very high speeds, exceeding 600 feet per second, the blades are made of steel milled from the solid edge of the wheel or set in a groove. Steel blades are used also for mixed pressure-turbines which take steam from the exhaust of reciprocating engines, but which have in addition jets of live steam acting on the first set of blades to provide for possible deficit of steam from the engine; these supplemental live steam jets are apt to loosen blades made of extruded metal or compositions.

Fig. 67 shows an alternative way of fixing extruded metal blades of marine turbines and other turbines with moderate peripheral velocity. A bar of steel is milled into a channel and sawed nearly across at intervals so that it can be bent into the arc of a circle and fit the edge of the wheel. It is punched through to receive the roots of the blades which are riveted over inside of the channel. After the blades are riveted into the channel bar, it is set in a groove

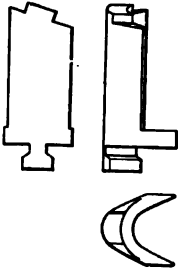


FIG. 66.

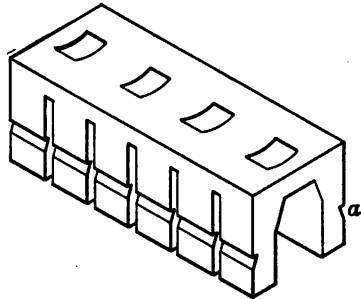


FIG. 67.

in the edge of the wheel as far as the nick at  $a$ , and the corner of the wheel is driven or calked down into the nick. The blades, which now stand out radially from the channel bar, are riveted to a shrouding at the outer ends as in Fig. 64.

The guides are set in the diaphragms that separate the pressure stages, and not being subjected to centrifugal force are more easily secured than the blades.

**Methods of Design.** — In the computations for design thus far stated, the various conditions for the solution have been assumed somewhat arbitrarily, including friction for nozzles, blades, and guides, disk friction and resistance, and mechanical efficiency. The friction for the nozzles is fairly well known and varies between narrow limits, but all the other factors are difficult to obtain and there is little published information concerning them, and consequently the direct and logical method thus far followed is involved in some uncertainty, though it can be controlled by so adjusting the successive factors that enter into the actual efficiency of the turbine, that the computed steam per horse-power per hour shall approximate to results of tests on turbines.

The most positive and valuable information that we have concerning steam turbines is the steam economy from tests either by the builders or other engineers. If, in addition, we have sufficient information concerning the construction of the turbines, the design may be based on the performance of turbines built and tested, thus following the accepted methods of steam engineering. Since the actual efficiency of the turbine is known, the uncertainties may be distributed among the several factors entering into it, so that none shall be much in error. Moreover, if the general design is good, minor variations will have only a secondary influence. For example, it is shown on page 51 that a simple impulse turbine may vary considerably from the speed of best efficiency without much effect on economy. Turbines with several pressure stages are even less sensitive to variations from the best conditions. For example, Fig. 68 shows the water rate for a 2500 K.W. five-stage turbo-generator, built by the

General Electric Company, operating with steam at 180 pounds by the gauge, 28-inch vacuum, and  $125^{\circ}$  superheat, under normal conditions, the governor in operation.

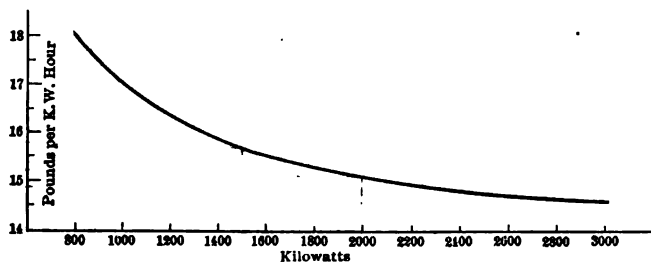


FIG. 68.

The same turbine, under the same conditions, but with six nozzles blocked open, showed the water rate indicated by Fig. 69 when run at various revolutions, the normal being 1500 per minute.

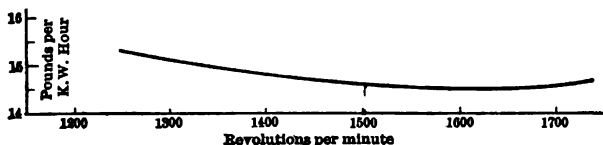


FIG. 69.

It may, therefore, be concluded that a turbine, which can show so little variation in performance when run under conditions so far removed from those which must have been taken for design, will give a high economy even though the details of design are varied considerably.

Of the several factors entering into the actual efficiency of a turbine, those which are most uncertain are the blade efficiency and the friction and resistance of the disk and attached parts. In fact, these factors can be separated only with difficulty and uncertainty. A method used by builders is to run a single pressure stage separately, or to build a single-stage turbine;

in either case the conditions for the stage under investigation can be simulated approximately. The product of the blade efficiency and the disk resistance can be thus investigated; at another time the disk resistance can be investigated separately, with a plane disk and with blades attached. Such investigations will perhaps be most useful in showing conditions to be avoided, such as rough surfaces and excessive blade lengths. The conditions may be expected to vary from stage to stage, but allowance for such influences cannot be included in a preliminary computation, which is as far as our work can be pressed.

We will therefore assume uniform conditions for all stages and will separate the factors under discussion somewhat arbitrarily. An error in the separation will have little, if any, effect on the determination of nozzle areas; and further, errors in nozzle areas will merely give somewhat unequal steam velocities from the jets, which have been shown to have but little influence. The most serious effect of an error will probably occur when too small a value is assigned to blade and guide friction, as it tends to restrict the areas of passages and choke the flow of steam. But, as already explained, the increase in blade length for a certain pressure stage depends principally on the angles, and only to a secondary extent on friction. For example, the table on page 140 shows that the increase in length of a blade due to friction is from 1 inch to 1.15, while the change of angle of the next guide, together with about half the frictional effect, increases the length to 1.51; the change in angle in this case is only  $4\frac{1}{2}^\circ$ . The table in question relates to a velocity-compound turbine with excessive velocities and, therefore, with large effect from friction (whether or not the factor is correctly selected), and so great an effect will not be found from friction in a pressure and velocity-compound turbine.

**Design of a Pressure and Velocity Compound Turbine.** — Let it be required to design a turbine having five pressure stages and two velocity stages for each pressure stage, with the following conditions:

Rated output, kilowatts . . . . .	2500
Maximum output, kilowatts . . . . .	3000
Pressure by the gauge, pounds . . . . .	180
Vacuum, inches mercury . . . . .	28
Superheating, degrees . . . . .	125
Revolutions per minute . . . . .	1500

The computations for this design are a combination of those for pressure compounding and of those for velocity compounding and consequently can be stated briefly, mainly in tabular form, with little detail of explanation. It will further be assumed that tests, on a turbine of similar arrangement, show that an actual water rate for such a turbine may be expected to be 14.5 pounds of superheated steam per kilowatt hour output. Also that tests on the turbine and generator show that the efficiency of transformation of energy, including all mechanical and electrical losses, is 0.9. Now a kilowatt is equivalent to about 1.34 horse-power, so that the steam must develop and apply to driving the turbine

$$1.34 \div 0.9 = 1.49$$

horse-power for each kilowatt output; consequently the turbine horse-power at maximum load will be

$$3000 \times 1.49 = 4470.$$

The steam per turbine horse-power will be

$$14.5 \div 1.49 = 9.72 \text{ pounds.}$$

The conditions for pressure and superheat will be found approximately on page 92 of the Entropy table at  $380^{\circ}$  and entropy 1.63; the vacuum corresponds nearly to  $102^{\circ}$ . The thermal properties are:

Entropy.	Temperature.	Pressure.	Superheat.	Heat contents.	Heat of liquid.
1.63	380	195.5	129.1	1271.4	...
1.63	102	1.0	...	910.4	70
				361.0	

The efficiency for Rankine's cycle by equation (6), page 35, is

$$e = 1 - \frac{C_2 - q_2}{C_1 - q_2} = 1 - \frac{910.4 - 70}{1271.4 - 70} = 0.300.$$

The steam per horse-power per hour for Rankine's cycle is by equation (7), page 36,

$$\frac{2545}{C_1 - C_2} = \frac{2545}{361.0} = 7.05 \text{ pounds.}$$

In the discussion of the overall heat factor, allowance should be made for radiation and leakage, though all the steam must pass through the first set of nozzles, which are the only entrance to the turbine. If two per cent is allowed for these losses then the actual steam consumption assumed for computation will be

$$9.72 \times .98 = 9.53 \text{ pounds.}$$

The overall heat factor used for heat distribution will be

$$7.05 \div 9.53 = .739;$$

and consequently the ratio for temperature distribution by the table on page 68 is about 1.05. We may now draw the diagram for the distribution factors as shown by Fig. 70. The factor for the first stage is 1.04, and, as shown on page 73, this is the ratio of the overall factor to the factor per stage. Consequently the heat factor per stage is

$$0.739 \div 1.04 = .711.$$

This heat factor per stage is made up of three factors: (1) the factor for the nozzle; (2) the blade efficiency, and (3) the factor for disk friction and resistance. If we take the friction factor for the nozzle to be  $y_n = 0.05$ , then the first factor is 0.95. The table on page 129 gives 0.83 for the efficiency for the two-wheel stage with a friction factor  $y_b = y_r = 0.16$ . The factor for disk resistance can be taken provisionally as 0.9. The product gives

$$0.95 \times 0.83 \times 0.9 = 0.71,$$

which coincides nearly with the value given above.

The difference in heat contents at  $380^\circ$  and  $102^\circ$  entropy 1.63, as found above, is 361.0 B.T.U., and, as there are five stages, this gives 72.2 B.T.U. for the preliminary heat assignment per stage at entropy 1.63. Taking the distribution factors from Fig. 70, the portions assigned for temperature distribution will be

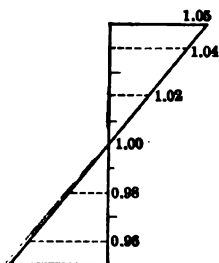


FIG. 70.

$$1.04 \times 72.2 = 75.1$$

$$1.02 \times 72.2 = 73.6$$

$$1.00 \times 72.2 = 72.2$$

$$0.98 \times 72.2 = 70.8$$

$$0.96 \times 72.2 = 69.3$$

$$\underline{361.0}$$

Subtracting these heat assignments from the initial heat contents, as shown in the following table, we have the heat contents at entropy 1.63 for which the temperatures and pressures are to be sought in the entropy table, as follows:

Nozzles.	Heat contents.	Temperature.	Pressure.
First . . . . .	{ 1271.4 75.1	380	195.5
Second . . . . .	{ 1196.3 73.6	317.5	86.5
Third . . . . .	{ 1122.7 72.2	257	33.6
Fourth . . . . .	{ 1050.5 70.8	201	11.8
Fifth . . . . .	{ 979.7 69.3 910.4	149.5 102	3.67 1.00

The pressures in the throats of the several sets of nozzles may be taken as 0.58 of the initial pressure, and the back-pressure for any set is the initial pressure for the next set, except that for the last set the back is the exhaust pressure; these pressures and the corresponding temperatures to the nearest half degree are given in the following table:

Nozzles.	Pressure.	Temperature.	Heat contents for entropy.	Entropy.	Heat contents.	Available heat.		Square roots.	Velocities.
						Adiabatic.	$\gamma = 0.05$ .		
First initial . . . .	195.5	380	1271.4	1.63	1271.4	...	...	...	...
throat . . . .	113.4	337	...	...	1219.6	51.8	49.2	7.01	1568
final . . . .	86.5	317.5	...	...	1196.3	75.1	71.4	8.45	1890
Second initial . . .	86.5	317.5	1218.0	1.66	1221.0	...	...	...	...
throat . . . .	50.2	281	...	...	1174.7	46.3	44.0	6.63	1483
final . . . .	33.6	257	...	...	1144.3	76.7	72.9	8.54	1911
Third initial . . . .	33.6	257	1164.6	1.69	1165.6	...	...	...	...
throat . . . .	19.5	227	...	...	1125.8	39.8	37.8	6.15	1376
final . . . .	11.8	201	...	...	1090.1	75.5	71.8	8.47	1895
Fourth initial . . .	11.8	201	1111.2	1.72	1109.9	...	...	...	...
throat . . . .	6.84	176	...	...	1073.7	36.2	34.4	5.87	1313
final . . . .	3.67	149.5	...	...	1034.4	75.5	71.8	8.47	1895
Fifth initial . . . .	3.67	149.5	1057.8	1.76	1058.8	...	...	...	...
throat . . . .	2.13	128.5	...	...	1026.0	32.8	31.2	5.59	1250
final . . . .	1.00	102	...	...	983.4	75.4	71.6	8.46	1892

Now the apparent available heat for adiabatic expansion from  $380^{\circ}$  to  $102^{\circ}$  at entropy 1.63, is divided into five equal parts of 72.2 B.T.U. each, and since the overall heat factor is 0.739, the heat assumed to be actually changed into work and applied to driving the turbine is

$$72.2 \times 0.739 = 53.4, \text{ B.T.U.}$$

so that the heat in the steam approaching the second set of nozzles is

$$1271.4 - 53.4 = 1218.0 \text{ B.T.U.}$$

and this is set down as the heat contents for the second set of nozzles. Subtracting the same amount from this quantity gives the heat contents for the third set of nozzles, and so on through the tables. If we were striving for the greatest precision possible, we would interpolate in the entropy table at  $317^{\circ}.5$  to find the entropy at which the heat contents should be 1218.3 B.T.U., but, even if somewhat greater precision were possible by such a method, it would not be of practical value, consequently we take

the nearest entropy column, which for the second stage is 1.66; in like manner the entropies for the other stages are determined and set down in the table.

For the first stage the heat contents at the throat-pressure in the nozzle is found at entropy 1.63 and temperature  $337^{\circ}$  to be 1219.6 B.T.U., and the heat contents at  $317^{\circ}.5$  at exit is found to be 1196.3 B.T.U. Consequently the apparent adiabatic available heat for developing velocity at the throat is

$$1271.4 - 1219.6 = 51.8 \text{ B.T.U.},$$

and the adiabatic available heat for velocity at exit is

$$1271.4 - 1196.3 = 75.1 \text{ B.T.U.}$$

In like manner the heat contents at throat and exit for the several nozzles are found from the entropy table, at the proper entropies and temperatures, and the adiabatic available heats are found by subtraction.

If our method of temperature distribution was absolute and the entropy table was sufficiently precise, the several available heats at the exit would all be equal to 75.1, as found in the computation for the first stage. In order to develop the effect of the distribution by our method, the computation for the exit velocity has been made for all the stages, and the greatest discrepancies appear to be 1.1 per cent. Again, if it were worth while, we might use these velocities for computing the nozzles and blade lengths for the several stages, but such refinement is of more than questionable value.

The available heat for velocity at throat and exit is assumed to be 0.95 of the adiabatic heat, thus making the friction factor for the nozzles  $\gamma_n = 0.05$ . In our previous designs in this chapter, the entrance to the nozzles has been assumed to be short, and no allowance for friction is made in computing velocity at the throat, but in this design, as also in the design for the Rateau turbine, the entrances are long and allowance for friction is here made for both throat and exit. In the table the square roots of

the available heats are set down, and these when multiplied by

$$\sqrt{2g \times 778} = 223.7$$

give the velocities for the throats and exits; the exit velocity for all nozzles may be taken as 1890 feet per second.

Now the water rate was 14.5 pounds per kilowatt hour, and the maximum load was 3000 kilowatts, so that the steam flow in pounds per second was

$$14.5 \times 3000 \div 3600 = 12.08 \text{ pounds.}$$

The total areas of the nozzles in square inches can be found by the equation

$$a = 144 wv \div V,$$

where  $w$  is the weight just given,  $v$  is the specific volume and  $V$  is the velocity in feet per second; this last is given for the various stages in the preceding table. The specific volumes are to be found by interpolation in the entropy table. For this purpose, we first find the heat contents of the steam at the throat and exit for each nozzle as set down in the table on page 175.

Nozzles.	Temperature.	Heat contents.	Specific volumes.	Total nozzle areas, sq. ins.	No. of nozzles.	Radial dimensions.
First initial . . . . .	380	1271.4	...	...	12	...
throat . . . . .	337	1222.2	4.34	4.81	...	0.504
final . . . . .	317.5	1200.0	5.34	4.91	...	0.515
Second initial . . . . .	317.5	1218.0	...	...	24	...
throat . . . . .	281	1174.0	8.55	10.03	...	0.527
final . . . . .	257	1146.6	12.09	11.13	...	0.584
Third initial . . . . .	257	1164.6	...	...	36	...
throat . . . . .	227	1126.8	19.81	25.1	...	0.877
final . . . . .	201	1093.2	31.21	28.8	...	1.006
Fourth initial . . . . .	201	1111.2	...	...	70	...
throat . . . . .	176	1076.8	51.2	68.0	...	1.007
final . . . . .	149.5	1039.8	89.7	82.6	...	1.223
Fifth initial . . . . .	149.5	1057.8	...	...	70	...
throat . . . . .	128.5	1026.6	148.9	208	...	3.080
final . . . . .	102	986.4	293.7	271	...	4.014

Thus, the heat contents in the steam approaching the *second* set of nozzles, as given in the fourth column of that table, is 1218.0, which quantity is repeated in the table in hand. From the same source we find that 44.0 B.T.U. are expended in producing velocity at the throat, and 71.4 B.T.U. have been assigned for producing velocity at the exit from each set of nozzles; consequently the heat contents at throat and exit are

$$1218.0 - 44.0 = 1174.0 \text{ B.T.U.}$$

$$1218.0 - 71.4 = 1146.6 \text{ B.T.U.,}$$

as set down for the second stage of the table. All the other quantities are found in like manner.

Now, turning to page 97 of the entropy table, the heat contents 1174.0 at temperature 281° is found to come between the entropies 1.65 and 1.66; the specific volumes corresponding to 1174.0 B.T.U. is found by interpolation, as illustrated on page 25, and in this manner all the specific volumes were determined. The total area for the throats of the second set of nozzles is therefore

$$a_t = 144 \times 12.08 \times 8.55 \div 1483 = 10.03 \text{ square inches,}$$

and the total area at exit is

$$a_e = 144 \times 12.08 \times 12.09 \div 1890 = 11.13 \text{ square inches,}$$

it having been decided to use 1890 feet per second for the exit velocity for all stages. The other areas are found by the same process and set down in the table.

In order to proceed with the design it is necessary to determine the diameter of the turbine, and the dimensions and arrangements of the nozzles, blades, and guides.

From the table for two-wheel velocity compounding, on page 129, we find that the peripheral speed of the turbine is 0.230 of the jet velocity, which gives

$$0.23 \times 1890 = 435 \text{ feet per second.}$$

The revolutions have been fixed at 1500 per minute and consequently the diameter will be

$$d = \frac{435 \times 60}{1500 \pi} = 5.54 \text{ feet or about } 66.5 \text{ inches.}$$

The blades and guides for the first three stages may have an axial width of 1.25 inches, and 1.50 inches for the two last stages; the pitches may be about half the axial width. The circumference of the pitch circle will be

$$66.5 \pi = 208.92 \text{ inches;}$$

consequently if there are 336 blades for the first three stages, and 280 for the last two, the pitches may be

First, second, and third stages,  $208.92 \div 336 = 0.621$  of an inch.

Fourth and fifth stages,  $208.92 \div 280 = 0.746$  of an inch.

The axial clearances may all be about one-eighth of an inch.

For simplicity of computation it will be assumed that the pitch of the nozzles will be four times the pitch of the blades, giving

First, second, and third stages,  $4 \times 0.621 = 2.484$  inches.

Fourth and fifth stages,  $4 \times 0.746 = 2.984$  inches.

Fig. 71 gives a developed cylindrical section of the nozzles, blades, and guides for the fourth and fifth stages; a section of the nozzles, blades, and guides for the three first stages would differ only in that the pitch and blade widths are smaller. The construction of blades and guides is like that explained on page 89. For further simplification the peripheral width of the blade edges has been taken as 0.04 of an inch for all blades and guides, and the plates separating the nozzles have been given a peripheral width of 0.16 of an inch, so that the first blade height shall be equal to the exit radial dimension of the nozzle, and that the exit height for a blade and the entrance height for a guide following shall be the same, etc. In practice

the thickness of the blade edge may be 0.02 of an inch, and the plates separating the nozzles are thicker.

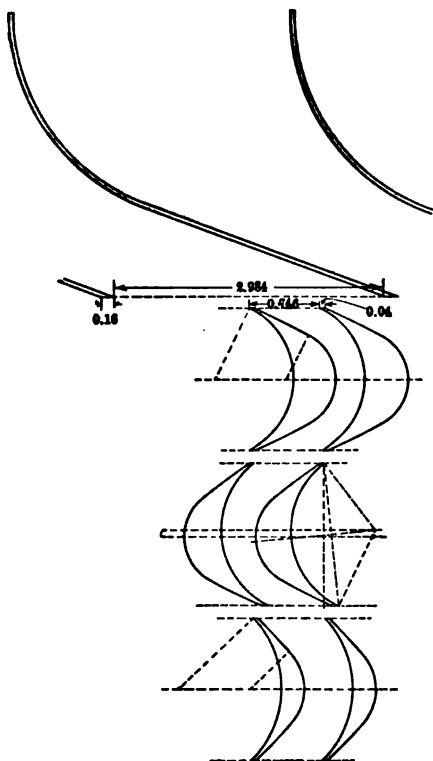


FIG. 71.

The construction of the turbine may be similar to that shown by Fig. 62, page 165, which represents a four-stage turbine made by the General Electric Company.

Since there are 280 blades on the wheels of the fourth and fifth stages, full peripheral admission will give 70 nozzles for those stages; this type can be constructed without any blank spaces in the diaphragms, the plates forming the nozzles being strong enough to permit it. The number of nozzles and their dimensions and arrangements will be determined as the design proceeds, and entered in the table on page 177.

The pitch of the nozzle being 2.984 inches, and the peripheral width of the dividing plate being 0.16 of an inch, the clear peripheral space is 2.824 inches. The sine of  $20^\circ$  is 0.342, consequently the net space between the plates is

$$2.824 \times 0.342 = 0.965 \text{ of an inch.}$$

But there are 70 nozzles, consequently the total net space of all the nozzles is

$$0.965 \times 70 = 67.55 \text{ inches.}$$

Dividing the total nozzle areas for the fourth and fifth stages,

on page 177, by this figure gives the radial dimensions of the nozzles at both throat and exit.

In like manner, since the pitch of the first three stage nozzles is 2.484 inches, and the peripheral width of the plates is 0.16 of an inch, the peripheral space is 2.324 inches, which, multiplied by the sine of  $20^\circ$ , gives

$$2.324 \times 0.342 = 0.795 \text{ of an inch}$$

for the net distance between plates.

By trial we may find that 36 nozzles will give a good arrangement for the third stage; the corresponding total net space is

$$0.795 \times 36 = 28.62 \text{ inches.}$$

Dividing the total areas at throat and exit for the third stage nozzles by this figure gives the radial dimension as set down. Again, taking 24 nozzles for the second stage, we get

$$0.795 \times 24 = 19.08 \text{ inches}$$

for the total net space, to be used for finding the radial dimensions of the nozzles of that stage as given in the table. Finally, taking 12 nozzles for the first stage, we have

$$0.795 \times 12 = 9.54 \text{ inches}$$

for the net space for finding the dimensions in the table for the first set of nozzles.

Coming now to the lengths of blades and guides, it will be remembered that, as shown on page 95, by the use of a constant peripheral width for the edges of blades and guides, and the relation shown between this quantity and the peripheral space occupied by the plates separating the nozzles, we have provided that the entrance length for the first set of blades is equal to the exit radial dimension of the nozzle, and exit length is the same as the entrance length of the following guides, and so on. It will therefore be sufficient to determine the several exit lengths of blades and guides.

Now the blade (and guide) lengths are directly proportional to the specific volumes, and inversely as the velocities and the net spaces between the blades (or guides); the net spaces are proportional to the sines of the angles.

In the design in hand the angles and velocities are the same for all the stages, so that it will be convenient to allow first for their influence and afterward for the effect of drying the steam, which latter effect will be of secondary importance.

	Exit angles.	Sines.	Velocity.	Factor.
Nozzle . . . . .	20°	0.342	1890	...
First blades . . . . .	25° 45'	0.434	1365	1.092
Guides . . . . .	25° 45'	0.434	909	1.638
Second blades . . . . .	45° 40'	0.715	508	1.780

The velocities are found by multiplying the jet velocity by the proper factors, from the table on page 129; thus the first-blade velocity is

$$1890 \times 0.722 = 1365.$$

The calculation for the exit of the first blades is

$$\frac{1890 \times 0.342}{1365 \times 0.434} = 1.092,$$

and the other factors are found in the same way. The blade and guide lengths are to be found (neglecting the effect of drying) by multiplying the exit radial nozzle dimensions for the various stages by the factors thus found, as set down in the table on following page.

In dealing with the effect of increase of specific volume due to transformation of kinetic energy into heat, we must consider, as explained on page 140, that the heat equivalent of the kinetic energy of a pound of steam having the velocity  $V$  is

$$V^2 \div 2 \times g 778 = 0.00001998 V^2.$$

Stage.	Exit length, approximate.	Temperature.	Specific volume.	Exit length, corrected for heating.
First Nozzle . . . . .	0.515	317.5	5.34	...
Blade . . . . .	0.562	...	5.45	0.575
Guide . . . . .	0.844	...	5.50	0.869
Blade . . . . .	0.917	...	5.51	0.946
Second Nozzle . . . . .	0.584	257	12.09	...
Blade . . . . .	0.638	...	12.18	0.643
Guide . . . . .	0.957	...	12.22	0.967
Blade . . . . .	1.040	...	12.23	1.052
Third Nozzle. . . . .	1.006	201	31.2	...
Blade . . . . .	1.098	...	31.4	1.105
Guide . . . . .	1.648	...	31.5	1.664
Blade . . . . .	1.790	...	31.6	1.813
Fourth Nozzle . . . . .	1.223	149.5	89.7	...
Blade . . . . .	1.335	...	90.4	1.345
Guide . . . . .	2.003	...	90.7	2.025
Blade . . . . .	2.177	...	90.8	2.204
Fifth Nozzle . . . . .	4.014	102	293.7	...
Blade . . . . .	4.38	...	296.0	4.42
Guide . . . . .	6.58	...	297.0	6.66
Blade . . . . .	7.15	...	297.3	7.24

The computation is therefore made, as in the following table, by finding the entrance and exit velocities by aid of the table on page 129 for  $y_b = 0.16$ , and computing the heat equivalents by the above equation; the difference in the heat equivalents is the heat gain on account of friction.

	Velocity.	Heat equivalent.	Heat gain.	Sum.
First wheel, entrance $V_1$ . . . . .	1490	44.4	...	...
exit $V_2$ . . . . .	1365	37.2	7.2	7.2
Guide, entrance $V_4$ . . . . .	992	19.7	...	...
exit $V_1'$ . . . . .	909	16.5	3.2	10.4
Second wheel, entrance $V_2'$ . . . . .	554	6.1	...	...
exit $V_3'$ . . . . .	508	5.1	1.0	11.4

Having the heat that the steam gains on account of frictional resistance, we may find the resultant heat contents, and therefrom the specific volume, by interpolation in the heat contents of the entropy table. For example, the heat contents at exit

from the first set of nozzles, by the table on page 177, is 1200.0, so that with 7.2 B.T.U. gain the contents at exit from the first blades will be 1207.2; interpolation in the entropy table, page 97, between entropies 1.64 and 1.65 at  $317^{\circ}.5$ , gives for the corresponding volume 5.45. But this interpolation is tedious, especially for half degrees, as in the case given. Nearly as great precision can be had by the following method: enter the entropy table with the nearest temperature ( $317^{\circ}$ ) and seek that entropy column which gives the nearest approximation to the specific volume at exit from the nozzles, as set down in the table on page 183. For the first nozzles the volume is 5.34 which leads to the entropy column 1.63. The tabular differences vary slowly and may therefore be taken somewhat at random. At the locality indicated, the tabular difference in heat contents between entropies 1.63 and 1.64 at  $317^{\circ}$  is 7.9, while the tabular difference for volumes is 0.121. The effect of an increase of 7.2 B.T.U. in heat contents is therefore to increase the specific volume by the amount computed in the proportion

$$7.9 : 7.2 :: 0.121 : 0.110,$$

and this, added to the volume 5.34, makes the volume 5.45, as set down in the table. In like manner all the specific volumes in the table are found, for the blade and guide exits. In the computation of the approximate blade lengths, in the table on page 183, no allowance was made for the increase in volume due to drying the steam; that allowance can now be made by increasing the lengths in proportion of the specific volumes just found to the specific volumes at exit from the nozzles.

For example, the corrected blade length at exit from the first set of the first stage is

$$5.34 : 5.46 :: 0.562 : 0.575.$$

Examination of the table will show that the allowance for transformation of kinetic energy into heat, on account of friction, is one or two per cent; some slight discrepancy is liable to appear in the computations, due to the number of approximations used in determining the lengths, but they are of no practical importance.

It will be remembered that an allowance of two per cent was made for radiation; properly the first stage nozzles, and blades, and guides should now be increased a like amount, since all steam enters the turbine through them. The following stages will be but little affected by this condition.

Having the blade lengths, the blades and guides may be laid out in a manner similar to that shown by Fig. 52, page 144.

**Modification.** — The design for the pressure and velocity-compound turbine just completed, and the design for a velocity-compound turbine on page 132, which is important mainly as an introduction to the latter, have been carried through in the most direct manner for sake of simplicity in instruction. There are important modifications that are likely to be met in practice.

To avoid delay on an unimportant trifle, the full rated gauge pressure of 180 pounds was taken for the design without comment; it might be desirable to allow a small drop of pressure from the steam main to the bowl of the nozzles, even though all the heat and kinetic energy will be conserved and carried into the jets.

The design was computed for the maximum load because, as shown by Fig. 68, page 170, turbines commonly give their best economy for that condition; at the rated normal load and still more at underloads, the pressures in the several chambers or stages decrease, and the pressure in the last stage is liable to be very small. There is always a chance that the rated vacuum may not be realized. In order to maintain work on the last stage wheels, the pressure in the preceding chamber is maintained at three or four pounds absolute for the maximum load. This condition can be had by taking one and a half or two pounds back-pressure for the design. At maximum load the steam from the last set of jets will have excessive velocity, which is equivalent to running that stage at a reduced velocity; but it has been shown that a large reduction in velocity for a simple wheel has a comparatively small effect on economy, and such an effect on one stage of a compound turbine will have but a small effect on the economy of the turbine.

Attention has been called to the necessity of avoiding choking of the steam by undue restriction of passages. Some designers are so much influenced by this condition, that they increase areas between blades and guides considerably beyond what our computation indicates, especially for the first set of blades which receives steam from the nozzles, and in the following guides.

For sake of establishing a simple relation of blade and guide angles, we have kept the angles for blades the same at entrance and exit, but have reduced the exit angles for guides. It probably is better to make exit angles smaller for both blades and guides, as is the habit of many builders, even though that will result in a higher peripheral velocity, because the blade lengths should increase progressively from the nozzle to the exit from the last wheel.

In order to simplify the careful computation made for blade lengths, the peripheral width of the blades and guides was taken to be 0.04 of an inch throughout, and further, with four blades per nozzle, the peripheral dimension of the plates separating the nozzles was taken as 0.16 of an inch. The thickness of the plate with 20° nozzle angle would be only

$$0.16 \times \sin 20^\circ = 0.16 \times 0.342 = 0.055 \text{ of an inch,}$$

corresponding nearly to plate gauge 17; for the first three stages the plate should be about 16 gauge, and for the last two stages about 13 gauge. The edges of the blades and guides are commonly made 0.02 of an inch thick, giving thus a decreasing peripheral width. It is doubtful if the refinements that we have followed in calculating blade lengths are necessary, and in any case, the exit lengths may be calculated and the entrance lengths made as great as the preceding exit lengths, to avoid choking the steam.

But, as already said, some designers prefer to increase the blade lengths beyond the proportion given by our calculation (or the equivalent), in which case the passages will not run full, and the action of the steam is liable to differ from what has been assumed in the theory commonly accepted.

Owing to the difficulty and expense of filing or otherwise forming rectangular nozzles, builders are inclined to use straight, reamed nozzles. In such case the blade lengths for the first set which takes steam from the nozzles must be at least equal to the diameter of the nozzle at exit; the spaces between the blades will consequently not be filled in that set. How the other sets of blades and guides should be treated is not clear, but we probably would not be far wrong if we treat them as shown in the table on page 183, without troubling about the increase of volume on account of friction.

Turbines are commonly given some method of applying steam to the lower stages to provide for overloads, but that condition will be treated in a future chapter.

**Comments.** — The theory of turbines and its application to design has been presented in the customary form with even more refinement than is customary in textbooks. Such refinement, if calling for much more time and trouble in design, may be of questionable propriety. It is, however, considered that the student should become familiar with such refinements, so that he may use them or not at discretion in his practice. It has also been considered best to maintain a reasonable degree of refinement which has led to our method of distributing temperature and pressure, and to consideration of the effect of the thickness of blade edges. But neither of those matters adds much to the difficulty of computation. The allowance for drying of steam and increase of volume does materially complicate calculation and is of doubtful value.

The factors which appear to be at once of the most importance and the greatest certainty are the steam consumption and the design of the nozzles. Having the steam consumption, the overall heat factor can be determined positively, and the distribution of temperature and pressure can be made with fair certainty, after which the design of the nozzles can be made directly.

It appears that there is considerable latitude in the assignment of angles for blades and guides, and in the determination of blade

lengths, without much sacrifice of economy. The greatest difficulty is met in trying to separate the blade efficiency from the heat and rotation losses, and concurrently in the assignment of values to the friction factors. The friction factor which is tacitly allowed to include losses from splattering and the resistance of blade edges is of importance in the determination of the proper peripheral speed of the turbine and in calculating blade lengths. But an examination of the tables on pages 129 to 131 shows that the angles and velocities vary slowly when the friction factor is changed, and since it has already been pointed out that the economy of a turbine is but little affected by changes in load and speed, we may conclude that the design of a turbine is insensitive to changes in friction factors. The tables just referred to were made up with constant values of the factor  $\gamma$  for each case, which seems to be the only simple way, because the factors appear to increase directly with the numerical value of the velocity. The effect of friction in velocity compounding in any case is found mainly in the early operations (such as the passage through the first set of blades), and it is scarcely worth while to vary the factor for the later operations, where the velocities are small.

**Heat and Rotation Losses.** — There are two ways of approaching the losses in a certain pressure stage due to friction and other resistance, and to other extraneous action. One way is that illustrated in this chapter, proceeding from the steam consumption to determine the overall heat factor and the temperature distribution factors, from which the heat factor per stage is found. This factor is considered to be the continued product of three factors, the nozzle factor (which is fairly well known), the blade efficiency, and the factor for heat and rotation losses; as said more than once, the separation of the last two factors appears uncertain.

Now the heat loss and rotation factor may have charged to it all influences external to the passage of steam through the blades and guides; these are (1) friction of steam on the disk and other parts of the wheel; (2) fanning action of blades not running

full of steam; (3) radial leakage due to centrifugal force or other influences; (4) shaft leakage. There may be other effects not recognized here. Some of these influences may be estimated separately and should receive attention, at any rate to see that they are not excessive.

**Stodola's Equation.** — The only published equation for estimating the effect of disk friction and fanning action of blades is that given by Stodola; as changed into English units by Dr. Lowenstein the equation is

$$\text{H.P.} = [0.02295 \alpha_1 D^{2.5} + 1.4346 \alpha_2 L^{1.25}] \left( \frac{V}{100} \right)^3 \gamma,$$

where  $D$  is the diameter of the wheel in feet,  $L$  is the blade length in inches,  $V$  is the peripheral speed in feet per second, and  $\gamma$  is the density of the medium. The factors are

$$\alpha_1 = 3.14 \quad \text{and} \quad \alpha_2 = 0.42;$$

the result is the horse-power absorbed. The blade resistance is for one row, all of which are idle; for two rows we may compute separately, and to allow for blades in action we may take a proportionate reduction in the term.

To apply to the first stage of the turbine for which a design is given, we have

$$D = 5.54, \quad L_1 = (.534 + .566) \div 2 = .550, \quad L_2 = (.855 + .930) \div 2 \\ = .893, \quad V = 435, \quad \gamma = 1 \div 5.5.$$

The specific volume 5.5 is taken from the table on page 183 as a sufficient approximation.

The term for the fanning of blades may be written

$$1.4346 \times 0.42 (\overline{.550}^{1.25} + \overline{.893}^{1.25}) = 0.81.$$

The term for disk friction has the value

$$0.02295 \times 3.14 \times \overline{5.54}^{2.5} = 5.20.$$

Now there are 336 blades on the wheel in this stage, and, as

there are 12 nozzles and four blades per nozzle, the allowance for blades in action gives for the term for fanning

$$\frac{336 - 48}{336} \times 0.81 = 0.69.$$

The horse-power consumed becomes, therefore,

$$(5.20 + 0.69) \overline{4.35^3} \div 5.5 = 88.$$

Now the turbine is expected to develop 4470 turbine horse-power and one-fifth of this is 894, so that the factor for rotation loss will be

$$88 \div 894 = 0.098$$

for the first stage. The factor for the other stages will be less as the density of the steam diminishes, even though the blade length increases.

**Shaft-Leakage.**—The clearance between the diaphragms separating the stages and the shaft may be estimated as 0.015 of an inch, and the leakage may be computed as for a nozzle, but with a large reduction on account of friction. If the shaft is 12 inches in diameter the perimeter will be 37.7 inches, which will give an area of

$$37.7 \times 0.015 = 0.56 \text{ of a square inch.}$$

The greatest leakage will be found for the first diaphragm, in which the second set of nozzles is located; the total area of these nozzles was computed to be 10.03 square inches at the throat, so that the leakage area is nearly six per cent of the nozzle area; but the flow through the nozzles is free, while that through the shaft clearance is restricted, and leakage may be estimated at one or two per cent.

**Friction Factors.** — There is little published information concerning the friction factors for blades and nozzles. It appears that the values increase with the velocity, which indicates that the friction increases at a higher power than the square of the velocity. The author has used the equation

$$y = 0.0001 V$$

1871

1871



in which  $V$  is the velocity through the blades or guides; this equation gives lower values that are sometimes used.

**Marine Type.** — As will be shown in a later chapter, the direct application of a steam turbine to marine propulsion requires the adaptation of propeller and turbine to give the most favorable combination. In general the propeller is given as small a diameter and as high revolutions as may be, without incurring disadvantageous conditions, and then the design of the turbine is made to conform, even though it requires comparatively low peripheral velocity and a large number of stages.

To illustrate the conditions for the marine type of pressure and velocity-compound turbine a computation will be given for a turbine with fourteen stages, the first with four rows of blades, the second to the sixth stages with three rows each, and the rest of the stages with two rows each; these low-pressure stages are set on a drum which, as explained later, is arranged to counter-balance the thrust of the propeller. The advantage of taking four rows of blades for the first stage is that the pressure and temperature are reduced in the first set of nozzles to a degree that is advantageous, when the large diameter of the casing is considered.

The general arrangement of the turbine is much like that shown by Fig. 72, which is proposed by the Fore River Ship-building Company for a scout cruiser; that turbine has sixteen stages of which eleven are on a drum. The reversing turbine is placed at the after end and has two stages only, one with five rows of blades and one with four.

The conditions for the computation are

Initial pressure absolute, pounds . . . . .	244
Final pressure absolute, pounds . . . . .	2
Steam per shaft horse-power per hour, pounds . . . . .	12.6

Since the drum counterbalances the propeller thrust, and all the bearings have forced oil lubrication, the mechanical efficiency

may be taken as 0.95, so that the steam per turbine horse-power per hour may be taken as

$$12.6 \times 0.95 = 12.0 \text{ pounds,}$$

no allowance being made for radiation, etc.

The entropy table at entropy 1.52 gives

Pressure 244.1	Temperature 399	Heat contents 1192.4
2	126	881.5
		310.9

The steam consumption for Rankine's cycle by equation (7), page 36, is

$$\frac{2545}{C_1 - C_2} = \frac{2545}{310.9} = 8.2 \text{ pounds.}$$

This leads to an overall heat factor of

$$8.2 \div 12.0 = 0.68,$$

for which the table on page 68 gives the temperature distribution factor 1.06.

The nozzle angles may be taken as  $20^\circ$  for all the stages, though some, if not all, of the drum stages are likely to have larger nozzle angles to avoid excessive blade lengths.

The pitch diameter will be assumed to be constant, and the jet velocities will be arranged to correspond. The friction factors will be taken as 0.14 for the first stage, 0.12 for the other intermediate stages, and 0.08 for the drum stages.

The tables on pages 129 to 131 give the ratios  $V : V_1$  of the peripheral speed to the jet velocity, as set down in the following table:

	$\gamma$ .	$\frac{V}{V_1}$ .	$\frac{V_1}{V}$ .	Ratios.	Squares.
First stage . . . . .	0.14	0.098	10.2	2.42	5.86
Second to sixth stages .	0.12	0.143	6.99	1.65	2.72
Seventh to fourteenth stages . . . . .	0.08	0.237	4.22	1.00	1.00

The reciprocal of the ratio  $V : V_1$  gives the ratio of the jet to the peripheral velocity, which latter must be constant, since the pitch diameter is the same for all wheels. The third column, therefore, shows that the first stage jet velocity is 10.2 times the peripheral velocity, that the second stage jet velocity is 6.99 times, and the drum stage jet velocity is 4.22 times the peripheral velocity. The several jet velocities are therefore in the ratios set down in the fourth column. But the heats required to produce these jet velocities are proportional to the squares, as given in the last column. Consequently we have that the heat assignment for the first stage should be 5.86 times as much as for a drum stage, and the heat assignment for the second to sixth stages should be 2.72 times as much.

Summing up we have

1 stage with 5.86	5.86
5 stages with 2.72	13.60
8 stages with 1.0	8.00
	<hr/>
	27.46

The apparent adiabatic available heat from  $399^\circ$  to  $126^\circ$  at entropy 1.52 is 310.9 B.T.U.; dividing by the above sum we have

$$310.9 \div 27.46 = 11.32 \text{ B.T.U.};$$

this is the assignment at that entropy for a drum stage. The first stage will have an assignment of

$$5.86 \times 11.32 = 66.33 \text{ B.T.U.},$$

and the second to sixth stages will have

$$2.72 \times 11.32 = 30.82 \text{ B.T.U.}$$

These quantities are entered in the table on page 195 for the several stages as required.

The next step is to find the temperature distribution factors by aid of Fig. 73, which is drawn with the ratio 1.06, as given on page 192, for the overall heat factor 0.68. The vertical line represents the adiabatic available heat at entropy 1.52, namely,

310.9 B.T.U. Starting at the top 66.33 B.T.U. are laid off for the first stage, and 30.82 B.T.U. for each of the five succeeding stages; the last eight stages have each 11.32 B.T.U. At the middle of each segment thus found, abscissæ are drawn on which the distribution factors are measured. These distribution factors are entered in the third column, on page 195, and enable us to

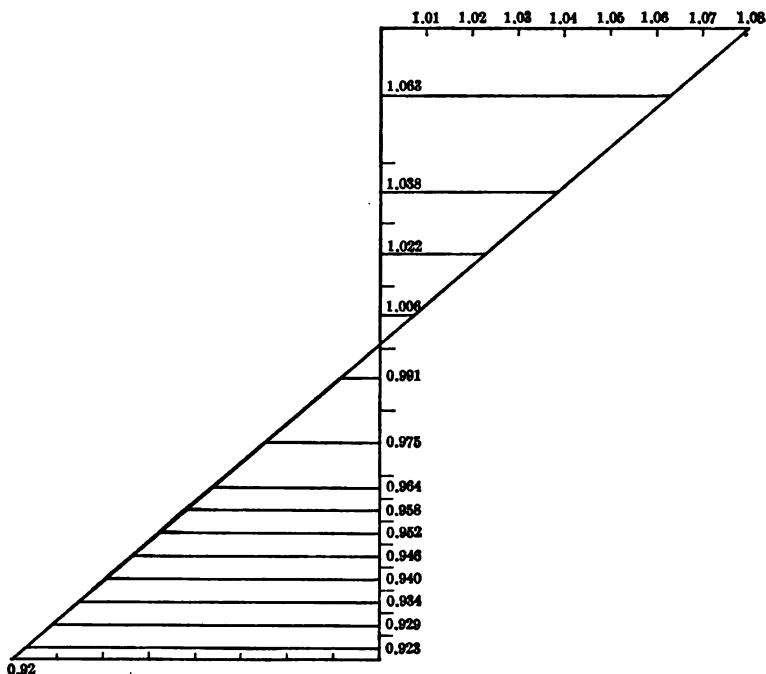


FIG. 73.

find the heat assignments at entropy 1.52, as set down in the fourth column. These assignments are subtracted, according to our custom, from the initial heat contents, giving for the first stage

$$1192.4 - 69.4 = 1123.0, \text{ etc.}$$

The table is arranged to include the computation for the throat of the first nozzle, where the pressure is

$$244.1 \times 0.58 = 141.6 \text{ pounds,}$$

and the corresponding temperature and heat contents are filled in, from the entropy table, on the line for the throat.

Stage.	Heat portion per stage.	Factor for temperature distribution.	Heat assignment.	Heat contents entropy = 1.52.	Temperature.	Absolute pressure.	Heat contents for volume.	Specific volume.	Total nozzle areas.
1	2	3	4	5	6	7	8	9	10
Initial Throat	...	...	...	1192.4	399	244.1	...	...	...
1	66.33	1.047	69.4	1123.0	329.5	102.3	1126.4	4.05	6.74
2	30.82	1.028	31.7	1091.3	299.5	66.5	1114.5	6.05	14.75
3	30.82	1.017	31.3	1060.0	271	42.5	1092.6	9.08	22.1
4	30.82	1.005	31.0	1029.0	244	26.8	1070.7	13.80	33.7
5	30.82	.993	30.6	998.4	218	16.5	1048.8	21.43	52.3
6	30.82	.981	30.2	968.2	193	10.0	1026.9	34.00	82.9
7	11.32	.973	11.0	957.2	184.5	8.3	1024.4	40.4	162
8	11.32	.968	10.9	946.3	176	6.9	1016.3	47.9	193
9	11.32	.964	10.9	935.4	167	5.6	1008.2	57.9	233
10	11.32	.960	10.9	924.5	158.5	4.6	1000.1	69.6	280
11	11.32	.955	10.8	913.7	150.5	3.7	992.0	83.1	334
12	11.32	.951	10.8	902.9	142	3.0	983.9	101.1	407
13	11.32	.947	10.7	892.2	134	2.5	975.8	122.3	492
14	11.32	.942	10.7	881.5	126	2.0	967.7	148.7	598

The columns of temperature and pressure can now be completed by interpolation in the entropy column 1.52 of the entropy table; as usual, half degrees only are taken, although for the drum stages this leads to some irregularity in the results, which are evident because the intervals are small.

One of the elements needed for computing the nozzle area is the specific volume, which can be found by interpolation in the entropy table after the heat contents in the steam is determined.

Taking first, the nozzles of the first stage, the adiabatic available heat at the throat is

$$1192.4 - 1148.3 = 44.1.$$

Assuming a friction factor of 0.05 for the nozzles, the heat actually changed into kinetic energy at the throat is

$$0.05 \times 44.1 = 41.9.$$

Subtracting this from the initial heat contents gives

$$1192.4 - 41.9 = 1150.5,$$

as set down in the eighth column of the table. For the exit of that nozzle we have

$$1192.4 - 0.95 \times 69.4 = 1126.4.$$

The heat assignment for the first stage of the turbine, as given in the table, is

$$66.33 \times 1.047 = 69.4,$$

the factor 1.047 representing the partial compensation for the stage heat losses, which comes from the fact that the succeeding stages can be computed at larger entropies. Our method of temperature distribution involves the assumption that a like compensation is found for all the stages, which is probably fair when the stages are all equal. When the stages are unequal our method has perhaps less validity and, in particular, the check calculation for the individual stages is less satisfactory. If the designer chooses, after accepting a series of intermediate temperatures and pressures he may compute the velocity for each stage separately, and also the specific volume for finding the nozzle areas; but such refinement is of doubtful value and will not be investigated here. We will take as the adiabatically available heat the following:

First stage,	$66.33 \times 1.047 = 69.4;$
Second to sixth,	$30.82 \times 1.047 = 32.3;$
Seventh to fourteenth,	$11.32 \times 1.047 = 11.9.$

In computing the heat contents at the exit from any nozzle, we must allow (1) for the heat changed into work in the preceding stages; (2) for the heat changed into kinetic energy in that nozzle.

For the first allowance we shall use the overall heat factor, giving

First stage,	$69.4 \times 0.68 = 47.2;$
Second to sixth,	$32.3 \times 0.68 = 21.9;$
Seventh to fourteenth,	$11.9 \times 0.68 = 8.1.$

For the second allowance we shall use the factor which comes from the friction factor  $\gamma_s = 0.05$ , giving

First stage,	$69.4 \times 0.95 = 66.0;$
Second to sixth,	$32.3 \times 0.95 = 30.7;$
Seventh to fourteenth,	$11.9 \times 0.95 = 11.3.$

To find the heat contents in the steam approaching the second set of nozzles, we have

$$1192.4 - 47.2 = 1145.2;$$

since 30.7 B.T.U. are changed into kinetic energy in those nozzles, the heat contents at exit will be

$$1145.2 - 30.7 = 1114.5,$$

as set down on the second line and in the eighth column of the table.

Proceeding now to the eleventh stage, we have for the heat changed into work in the preceding stages

$$47.2 + 5 \times 21.9 + 4 \times 8.1 = 189.1 \text{ B.T.U.},$$

and for the heat changed into kinetic energy in those nozzles 11.3, so that the heat contents at exit is

$$1192.4 - 189.1 - 11.3 = 992.0,$$

as set down for the eleventh stage in column 8.

The specific volumes given in the ninth column of the table are found by interpolation in the entropy table at the given temperatures.

The actual available heats for producing velocity in the nozzles have already been stated, and can be used for computing velocities as follows:

First stage, throat,	$V_t = 223.7 \sqrt{41.9} = 1446;$
First stage, exit,	$V_1 = 223.7 \sqrt{66.0} = 1818;$
Second to sixth stages,	$V_1 = 223.7 \sqrt{30.7} = 1240;$
Seventh to fourteenth stages,	$V_1 = 223.7 \sqrt{11.3} = 752.$

Let us now assign 6000 for the shaft horse-power of one turbine, then the steam per second will be

$$\frac{12.6 \times 6000}{60 \times 60} = 21 \text{ pounds.}$$

The total area in square inches for a set of nozzles is given by the equation

$$a = 144 wv \div V_1,$$

where  $w$  is the pounds of steam per second,  $v$  is the specific volume in cubic feet, and  $V_1$  is the jet velocity in feet per second; for the first set of nozzles we have also a calculation for the throat area, using the throat velocity  $V_t$ . For example, the total exit area for the first set of nozzles is

$$144 \times 21 \times 4.05 \div 1818 = 6.74 \text{ square inches.}$$

After the nozzle areas have been found, the design of the nozzles, blades, and guides should proceed as in the design for the five stage turbine, on page 171, differing only in that there are many pressure stages and a varying number of rows of blades in those stages.

## CHAPTER VIII

### REACTION TURBINES

THE essential feature of a reaction turbine is a fall of pressure, and a consequent increase of velocity, in the passages among the moving blades of the turbine. Since such wheels commonly are affected by impulse also, they are sometimes called impulse-reaction turbines, but the shorter name need not lead to confusion. In consequence of the feature named, the relative velocity of the steam at exit from the blades is greater than the relative velocity at entrance. Another consequence is that steam leaks past the ends of the blades, which are usually open, and there is leakage past the inner ends of the guides, which are also open; this feature is shown by Fig. 74.

The reaction turbine is always made compound with a large number of stages, one set of fixed guides and the following set of movable blades being counted as a stage. In consequence the exit pressure from the guides or blades is only a little less than the entrance pressure, and the passages between the guides or blades are always converging. As the pressure diminishes, the area for the passage of steam increases rapidly, and in order to avoid excessive blade lengths the turbine is commonly made in sections with increasing diameter. Fig. 75 gives the section of a Parsons turbine.

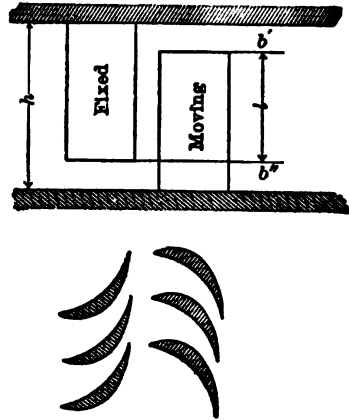


FIG. 74.

Steam is admitted at *A* and passes in succession through the stages of the high-pressure cylinder, and thence through the passage at *E* to the stages of the intermediate cylinder; after passing through the intermediate stages, it passes through *G* to the low-pressure stages and finally by *B* to the condenser. This figure is to be considered as a diagram, the minor features being too small to show; in particular the several stages on a cylinder are usually made in groups which have increasing blade height.

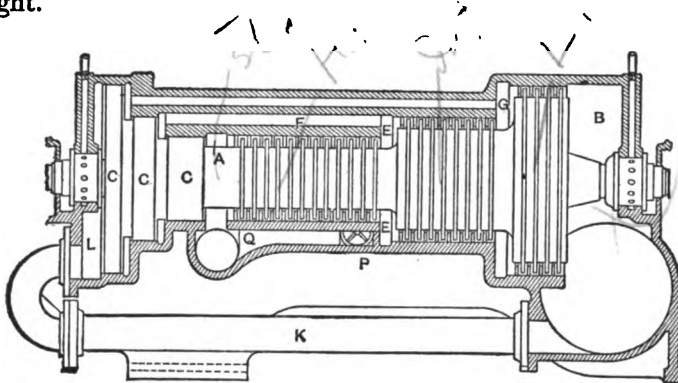


FIG. 75.

It will appear later that there is little, if any, axial thrust on the blades of the turbines and consequently, if there were no change in the diameter of the rotor, there would be no appreciable end thrust. The steam pressure on the effective areas presented by the several cylinders is counterbalanced by dummy cylinders, each of which has a diameter nearly equal to the pitch diameter of the corresponding bladed cylinder, and passages are carried either within or without the turbine casing to equalize the pressure. Leakage past the dummy cylinders is checked by a labyrinth packing, to be described hereafter.

**Velocity Diagram.**— The guides and blades follow alternately in close succession, having only the necessary axial clearance; the kinetic energy, due to the absolute exit velocity from a given set of blades, is not lost but is available in the next set of

guides. At the end of a section or cylinder this kinetic energy is changed into heat, which is available for the next cylinder with little loss of efficiency; at the end of the low-pressure cylinder the kinetic energy is thrown into the exhaust; but the total loss of efficiency is small, and it is doubtful if any account need be taken of it. The first set of guides for a given cylinder should strictly be specially shaped, as they do not receive steam from blades of a preceding stage; also more heat should be assigned to producing velocity in them; but refinements of this nature are not observed.

Since the exit absolute velocity of the steam from the blades is applied to driving it into the next set of guides, there appears to be no reason for restricting the velocity of whirl, it being necessary only to give the guides the proper angle at entrance, except that high velocities of steam among the blades and guides is, of course, to be avoided, since the friction increases with the square of the velocity.

In the preceding types of turbines the steam issues from nozzles with more or less resemblance to the isolated jet, which forms the basis of the theoretical discussion, but in this type the annular passage between the rotor and the case everywhere runs full of steam, which expands as it goes, and meets with the guides and blades which give and extract a whirling motion. There is, therefore, a resemblance to the passage of a stream among various obstacles, and the real action may involve hydrodynamic conceptions of great complexity. There is further discouragement from the usual theoretical treatment, because it is stated that the Parsons Company do not use the results of such a discussion in design, but rather depend on certain empirical equations, which, though crude, are so controlled by comparison with tests as to give reliable results with little labor.

Nevertheless, it is desirable to develop the customary velocity diagram and show how it is used for design.

The forms of the blades and guides for a certain stage are habitually the same, so that the velocity diagram takes the form shown by Fig. 76. The exit guide angle is  $\alpha$  and  $ab = V_1$  is the



and

$$bg = V_1 \cos \alpha, \quad ag = V_1 \sin \alpha;$$

$$\therefore \tan \beta = \frac{ag}{bg} = \frac{ag}{b_1 - V} = \frac{V_1 \sin \alpha}{V_1 (\cos \alpha - n)} = \frac{\sin \alpha}{\cos \alpha - n}. \quad (1)$$

Since there is appreciable kinetic energy in the steam entering the guides, it is necessary to retain the term  $V_2^2 \div 2g$  in equation (1), page 15, and consequently instead of equation (8), page 18, we have

$$\frac{V_1^2}{2g} = \frac{V_2^2}{2g} + 778 (C_1 - C_2), \quad \dots \quad (2)$$

where  $C_1$  and  $C_2$  are the heat contents for the conditions at entering and leaving the guides. But from Fig. 76

$$V_2^2 = ag^2 + gc^2 = V_1^2 \sin^2 \alpha + V_1^2 (\cos \alpha - n)^2. \quad \dots \quad (3)$$

Substituting in equation (2) and solving to the heat contents

$$C_1 - C_2 = \frac{V_1^2}{2g \times 778} \{1 - \sin^2 \alpha - (\cos \alpha - n)^2\}. \quad \dots \quad (4)$$

Equation (4) gives the adiabatic heat assignment for the guides; the assignment per stage is, of course, twice as great. Having the heat per stage we are in condition to determine the number of stages, the steam velocity, and finally the blade lengths.

**Axial Thrust.** — As there is no change of the velocity of flow indicated by Fig. 76 then, without friction, there would be no axial thrust; there will be little, if any, thrust in any case.

**Choice of Conditions.** — The most authoritative statement of conditions to be chosen for the design of a Parsons turbine is found in a valuable paper by Mr. E. M. Speakman,\* from which liberal quotations are made here and in the chapter on marine propulsion.

In general, the design of a reaction turbine may begin with the selection of the peripheral speed of the rotor at the pitch surface. Where there are two or more cylinders with varying diameters, the peripheral speed for the largest and smallest will

\* Trans. Inst. Eng. & Shipbld., Scot., 1905-6.

both be selected, which will determine the relation of the diameters. For electrical work there is considerable latitude in design, because both turbines and generators may be varied; even so, certain conditions have been found convenient as giving a fair weight to cost and efficiency for various powers. For marine work the conditions are closely limited by the design of the propeller and the location of the turbine, unless the revolutions of the turbine by the intervention of gearing, electrical transformations, or other means can be made greater than for the propeller.

Speakman gives the following tables of conditions for electrical and marine turbines:

PARSONS TURBINES — ELECTRICAL WORK

Normal output, kilowatts.	Peripheral speed, feet per second.		Number of stages.	Revolutions per minute.
	First expansion.	Last expansion.		
5000	135	330	70	750
3500	138	280	75	1200
2500	125	300	84	1360
1500	125	360	72	1500
1000	125	250	80	1800
750	125	260	77	2000
500	120	285	60	3000
250	100	210	72	3000
75	100	200	48	4000

The maximum speed for Parsons turbines at the time of his paper was 176 for high-pressure blades and 375 for low-pressure blades for electrical work.

PARSONS TURBINES — MARINE WORK

Type of vessel.	Peripheral speed, feet per second.		Ratio of velocities, blades to steam.	Number of shafts.
	H.P.	L.P.		
High speed mail steamers. . . . .	70-80	110-130	.45-.5	4
Intermediate steamers . . . . .	80-90	110-135	.47-.5	3 or 4
Channel steamers . . . . .	90-105	120-150	.37-.47	3
Battleships and large cruisers . . . . .	85-100	115-135	.48-.52	4
Small cruisers . . . . .	105-120	130-135	.47-.5	3 or 4
Torpedo craft. . . . .	110-130	160-210	.47-.51	3 or 4

A very common ratio of peripheral speed to the velocity of the steam for electrical work is 0.6.

It is evident that turbines of the Parsons marine type are frequently designed to run at less than the theoretical speed, as indicated by the discussion on page 203; for while a geometrical construction can be made for the ratios of  $V$  to  $V_1$  (i.e., of peripheral to steam speed) given in the above table, an attempt to draw blade sections with the small values of the entrance blade angle  $\beta$ , which such constructions would indicate, will show that a ratio much less than

$$V \div V_1 = n = 0.5$$

is impractical.

Having selected the peripheral speed and the ratio to the steam velocity, it is necessary to determine the exit angle  $\alpha$  of Fig. 76 and construct or compute the angle  $\beta$  for the entrance. Then the heat per stage may be calculated by aid of equation (4); this is the heat actually changed into kinetic energy. The heat allowed per stage, allowing for friction, tip leakage, etc., will be greater than that so calculated, and can be found by dividing by the overall heat factor.

Mr. Parsons at one time said, "For all practical purposes, while steam is traversing each set (of blades) as shown, it behaves like an incompressible fluid, just like water would do, as the expansion is very small at each set. The frictional losses and eddy-making losses would be practically identical, within small limits, with what they would be with water, and the actual forces would be in proportion to the density of the medium. . . . In the turbine blades themselves, the efficiency is between 70 and 80 per cent."

The overall heat factor is probably somewhat smaller than the efficiencies just quoted, and must be related to the steam consumption, as in previous work, and as will appear in the calculation for a design.

In view of the fact that but little is known concerning the blade angles and conformations for Parsons turbines, Fig. 77 is reproduced from Speakman's paper, in which it is given as the

typical arrangement for marine turbines. The peripheral speed is here half the steam velocity; the exit angle is  $22^{\circ}\frac{1}{2}$  and the entrance angle, which is drawn with a tangent to the *back of the blade*, is  $42^{\circ}$  as indicated both by construction and as computed by equation (1). It will be seen that the exit edge is given an appreciable thickness and has parallel sides, well adapted to guide the steam. The thickness is about ~~one-third the clear~~ space between the blades when measured on a line normal to

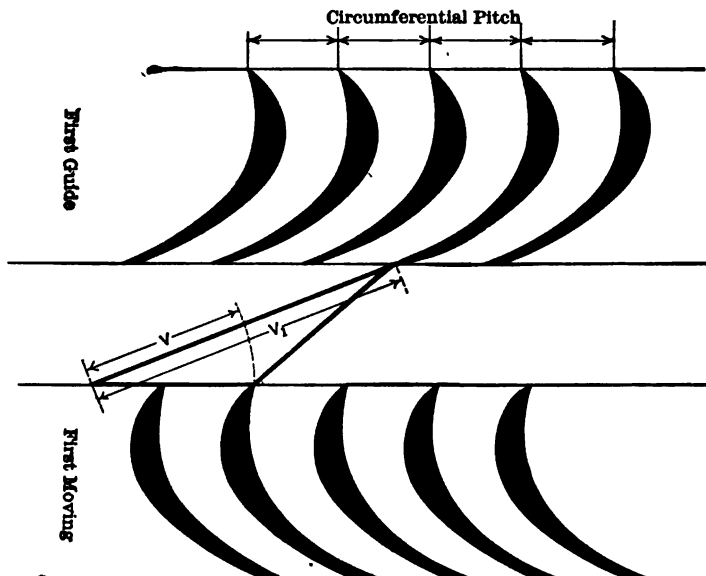


FIG. 77.

the blade contour at the tip. The blades of such turbines are commonly tapered to a thin edge at exit, so that the exit angle and the thickness of metal to be allowed in determining clear space become indefinite.

The pitch diameter of the rotor in feet is

$$d = \frac{60V}{\pi R}, \quad . . . . . (5)$$

where  $V$  is the peripheral speed in feet per second, and  $R$  is the

number of revolutions per minute; this applies to both the smallest and largest diameters. The intermediate diameter may be made approximately a mean proportional between the largest and smallest, when there are three cylinders.

The work may be divided about equally among the cylinders; consequently the number of rows for a cylinder may be made inversely proportional to the heat assignment per stage. The heat assignment per stage is proportional to the square of the steam velocity.

The net area in square inches for the passage of steam may be computed by the equation

$$a = 144 wv \div V_1, \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where  $w$  is the weight of steam in pounds per second,  $v$  is the specific volume, and  $V_1$  is the exit velocity in feet per second.

The net distance between the tip of a blade and the back of the next blade is equal to the pitch, multiplied by the sine of the angle, less the thickness of the metal measured on a normal to the blade contour at the tip. The net distance, therefore, depends on the blade angle and contour; the blade thickness will commonly be about one-third the net distance. Consequently the net area will be about

$$\frac{3}{4} \pi d \sin \alpha \cdot h = a,$$

where  $d$  is the diameter of the pitch surface in inches, and  $h$  is the blade height in inches. We have, therefore,

$$h = \frac{4a}{3\pi d \sin \alpha} = \frac{144 \times 4 \times wv}{3\pi d V_1 \sin \alpha} \quad . \quad . \quad . \quad . \quad (7)$$

If the thickness of the metal bears a different ratio to the clear space, allowance must be made taking the proper ratio in place of 3 : 4.

The blade height is properly the radial distance between the rotor and the case, not allowing for the tip clearance.

The ratio of blade height to the pitch diameter should not be less than 3 per cent, nor more than 15 per cent; because, in the

first place, the tip leakage is likely to be excessive, and, in the second place, the blades lack strength and rigidity.

Dimensions and proportions for blades and accessories, as recommended by Speakman, are given in Fig. 78 and the annexed table.

The axial clearances may be slightly reduced at discretion when there is special reason for reducing the length of the turbine. The standard dimensions have been determined mainly for convenience in overhauling.

The leakage over the tips of the blade (and under the guides) affects the condition and specific volume of the steam, and therefore the blade length. It appears as though special allowance should be made for the tip clearance, which bears a varying ratio to the blade length; but the information for this purpose is not at hand.

Speakman gives a diagram of allowable tip clearances which can be represented by the equation

$$\left. \begin{array}{l} \text{Clearance} \\ \text{in inches} \end{array} \right\} = 0.01 + 0.008 \text{ diameter in feet.}$$

For convenience in manufacture the blade heights are kept constant for several rows of blades. A common arrangement, when there are three cylinders, is to use only three blade heights on each cylinder. The blades are set with the mid length at the pitch surface, so that as the blade length increases the diameter of the rotor is correspondingly reduced. In marine practice, on the other hand, the rotor diameter for a certain turbine is kept constant, and consequently the pitch surface increases in diameter as the blade length grows. A group of blades, which have the same heights, is called a barrel; so that a typical arrangement is to have three cylinders, with three barrels on each, so that there are nine in all.

The material of which blades are made is said by Speakman to be a mixture of 16 parts copper and 3 parts tin; alloys containing zinc are unreliable at high temperatures. He recommends highly a composition of 80 parts copper and 20 parts nickel.

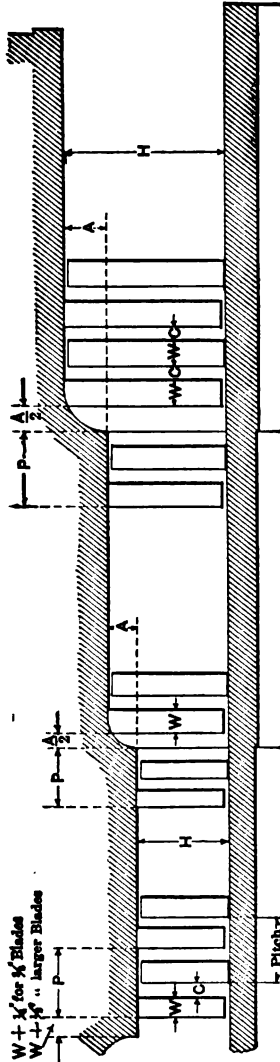


FIG. 78.

Height (H)	1"	2"	3"	4"	6"	8"	10"	12"	15"	18"	21"	24"	30"
Width (W)	$\frac{1}{16}$ "	$\frac{1}{8}$ "	$\frac{1}{4}$ "	$\frac{1}{2}$ "	$\frac{3}{8}$ "	$\frac{1}{2}$ "	$\frac{5}{8}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{2}$ "	$1\frac{3}{4}$ "
Pitch (P)	$\frac{1}{16}$ "	$\frac{1}{8}$ "	$\frac{1}{4}$ "	$\frac{1}{2}$ "	$\frac{3}{8}$ "	$\frac{1}{2}$ "	$\frac{5}{8}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{2}$ "	$1\frac{3}{4}$ "
Axial clearance (C)	$\frac{1}{16}$ "	$\frac{1}{8}$ "	$\frac{1}{4}$ "	$\frac{1}{2}$ "	$\frac{3}{8}$ "	$\frac{1}{2}$ "	$\frac{5}{8}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{1}{2}$ "	$1\frac{3}{4}$ "

Note. — While the above represents general practice, it is obvious that such a table is largely arbitrary.

The greatest difficulty in the operation of turbines has been the stripping of blades. Stripping may be charged to various conditions, all of which are preventable, such as (1) bad workmanship in setting the blades; (2) defective blade material; (3) excessive cylinder distortion; (4) whipping of turbine spindles; (5) wear of bearings, and (6) introduction of extraneous materials like water or grit. The first two conditions may be guarded against by inspection; the third and fourth show bad design or bad balancing; the last two conditions show poor operation. It is asserted that blades have been stripped without fouling, by small vibrations of high frequency.

The bearing pressure of a Parsons turbine is due to the weight of the spindle only. Speakman's rule is that the product of the pressure in pounds per square inch, by the velocity of rubbing in feet per second, should not exceed 2500 to 2700. In marine practice this allows 90 pounds per square inch at 30 feet per second. For stationary work, a common rule is to allow 50 pounds per square inch at 50 feet per second. Forced lubrication and oil cooling are demanded, with special kinds of oil. If possible the oil temperature should not exceed  $150^{\circ}\text{F.}$ , though a temperature as high as  $190^{\circ}\text{F.}$ , has been used successfully.

The casings, especially for marine turbines, are very large, and are ribbed for stiffness; unless designed with discretion, they are liable to be distorted when hot, because the temperature may vary from  $400^{\circ}\text{F.}$  to  $100^{\circ}\text{F.}$  in a distance of six or eight feet.

**Blading.** — The blades for a Parsons turbine are made of brass rods having the proper section, which are cut to the right length and grooved or crimped transversely at the fixed ends. The outer ends may be thinned so that there will be less danger of stripping, if they should happen to touch the casing. A groove is cut in the rotor for each row of blades, a trifle wider than the axial width of the blade. Filling pieces of soft brass are provided, a trifle longer than the depth of the groove, and shaped like the space between two blades, as shown by Fig. 77, page 206. The workman sets several blades and filling pieces in the groove,

one after the other, and then forces them up against those already secured; the filling pieces force the blades into correct positions, and any slight irregularity can be corrected by gauging them after they are secured. A calking tool is then introduced between the blades and the filling or calking pieces are upset or calked solidly into the groove; they also grip the ends of the blades by the transverse grooves. The guides are set in the same way; there is a special construction at the longitudinal joints of the case.

If the blades are long enough to lack rigidity they are stiffened by lashing them to a peripheral wire near the ends. For this purpose a square notch is cut in the thick or leading edge of each blade before it is set; a square wire is bent round and set into the notches of a row of blades; then a small, tough wire is wound round both blades and wire, so that they are lashed firmly together; afterwards the blades, wire, and lashing are soldered together with hard solder. Very long blades may be lashed also at mid height.

This standard method of blading is kept by the Parsons Company, though a number of variations have been introduced by the lessees of the Parsons patents. The chief criticism appears to be that the method is slow and only a few men can work on a given rotor or case at the same time. To meet this difficulty, various methods have been devised to assemble the blades and calking pieces in segments, before they are set into the rotor or casing.

The blading of turbines of this type by the Willans Company is represented by Fig. 79 and Fig. 80. The blades, which have the proper section in the body, have a flat tang stamped on the fixed end. These tangs are forced into slots in the foundation ring *F* and bent over, as shown by Fig. 80. A shallow channel shroud *S* is bent over the ends of the blades, and tits which protrude through holes in the shroud are riveted over. The segments are secured in the rotor and casing by

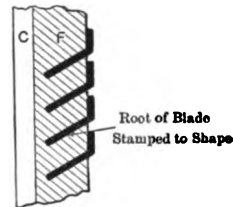


FIG. 79.

calking rings C. It is claimed that tip leakage is reduced by the shrouding.

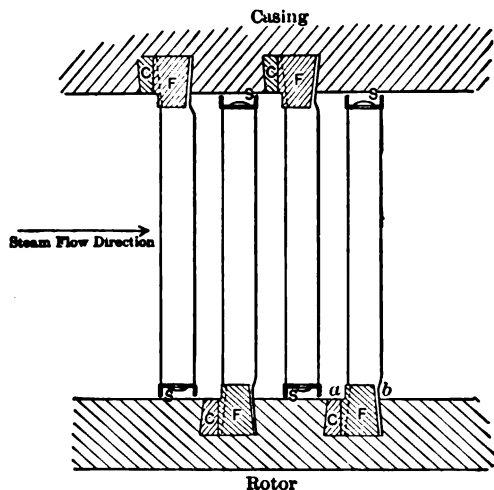


FIG. 80.

Fig. 81 shows the lacing of the blades of the turbine built by the Brown-Boveri Company. A round wire is rove through

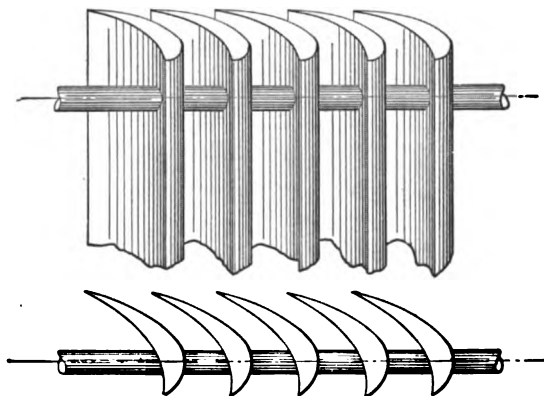


FIG. 81.

holes near the end of the blades and soldered directly to them. This wire has an iron core for strength, and a copper sheathing to facilitate soldering; the iron core has the same coefficient of

expansion as the iron rotor. Fig. 82 shows the appearance of the blades as set in the rotor, and is characteristic of the Parsons type without shrouding.

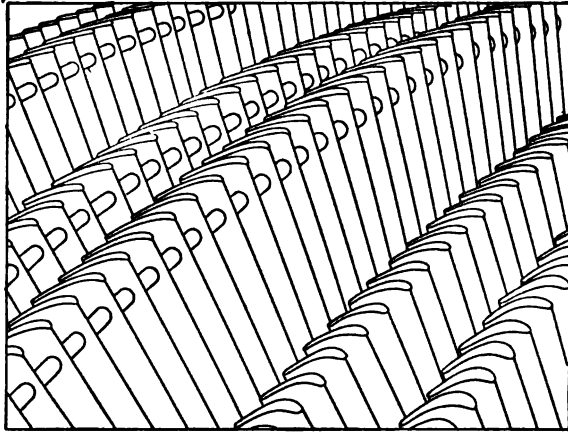


FIG. 82.

The Westinghouse Company have attempted to insure against stripping blades, by the arrangement shown in Fig. 83a. The blades are set in a foundation ring, which is set in a dovetailed groove and secured by a dovetailed strip, of which the details need not receive further attention. There are steady-strips near the tops of the blades, which are shaped so as to avoid interference of flow of the steam. The special feature is the employment of an adjustable ring *R* to which the guides are secured. This ring, which carries two sets of guides, has shoulders which prevent it from advancing too near the rotor, but it is held against the shoulders by flexible springs only. If for any reason the blades touch the ring *R*, it can bear on them only with the pressure due to the springs, which is not enough to strip the blades; it is expected that in such case the blades will wear off at the ends till they cease to bear. In much the same way, if the guides bear against the rotor, it is only with the pressure due to the springs, and they also are expected to wear until they cease to bear.

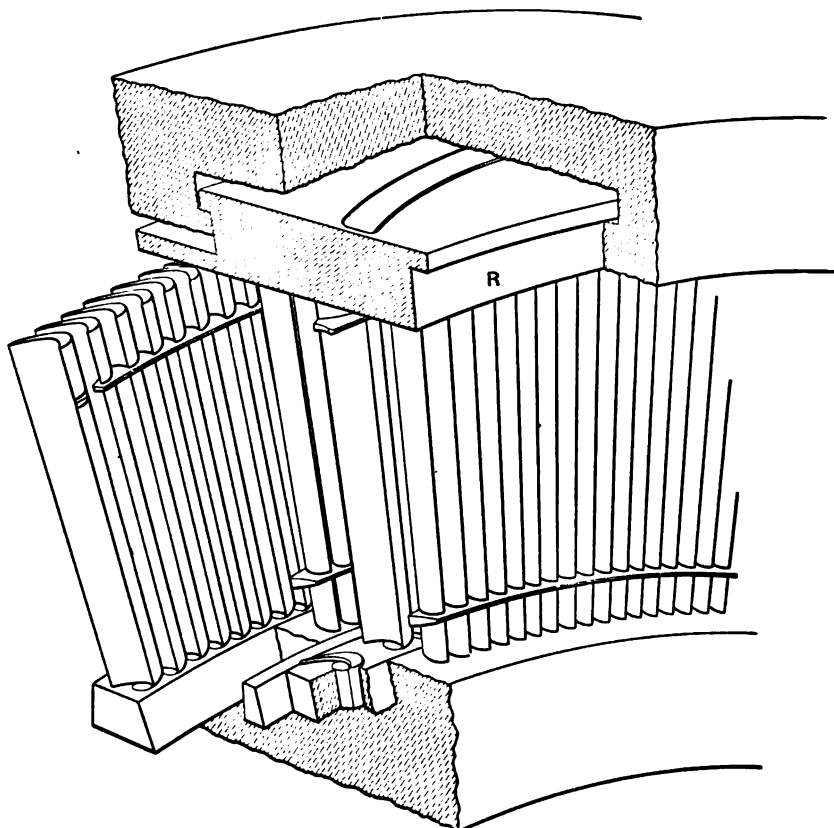


FIG. 83A.

**Design for a Reaction Turbine.** — Let the following conditions and data be assumed for the design of an impulse-and-reaction turbine:

Brake horse-power . . . . .	8500
Gauge pressure, initial . . . . .	150
Vacuum, inches of mercury . . . . .	28
Peripheral speed, small cylinder, feet per second . . . . .	135
Peripheral speed, large cylinder, feet per second . . . . .	330
Exit blade angle, degrees . . . . .	22½
Ratio of peripheral speed to steam velocity . . . . .	0.6
Revolutions per minute . . . . .	750
Overall heat factor . . . . .	0.75 ✓
Mechanical efficiency . . . . .	0.95 ✓
Radiation and leakage . . . . .	0.05

The conditions lead to the following thermal properties, with atmospheric pressure of 14.7 pounds:

Pressure. absolute.	Tempera- ture.	Entropy.	Heat contents.	Heat of liquid.
164.7 1	366 102	1.56 1.56	1193.3 871.1	...
			322.2	70

The efficiency for Rankine's cycle is

$$e = 1 - \frac{C_2 - q_2}{C_1 - q_2} = 1 - \frac{871.1 - 70}{1193.3 - 70} = 0.287.$$

The steam per horse-power per hour for Rankine's cycle is

$$W_R = \frac{2545}{C_1 - C_2} = \frac{2545}{322.2} = 7.89 \text{ pounds. } \checkmark$$

With an overall heat factor of 0.75, the steam per turbine horse-power is

$$7.89 \div 0.75 = 10.5 \text{ pounds. } \checkmark$$

The combined factor for mechanical friction and leakage is

$$0.95 (1 - 0.05) = 0.902, \checkmark$$

consequently the steam per shaft horse-power per hour is

$$10.5 \div 0.902 = 11.65 \text{ pounds. } \checkmark$$

This turbine may fairly be rated as having 5000 K.W. when connected to an electric generator, for the electric losses may be represented by a factor 0.95, and the ratio of the K.W. to a horse-power is about 1.34, so that we should expect

$$8500 \times 0.95 \div 1.34 = 6000 \text{ K.W. }$$

output at the designed condition; and this corresponds to 20 per cent more than the rating. The peripheral velocities and revolutions per minute are selected from page 209.

With the assigned exit blade angle and ratio of peripheral

speed to steam velocity, we may construct a diagram like Fig. 76, or may compute the entrance angle by the equation (1), page 203.

$$\tan \beta = \frac{\sin 22^{\circ}\frac{1}{2}}{\cos 22^{\circ}\frac{1}{2} - 0.6} = \frac{0.3827}{0.9239 - 0.6} = 1.185.$$

$$\beta = 49^{\circ} 50'.$$

The actual heat required to accelerate the steam in the guides or blades is by equation (4)

$$\frac{V_1^2}{2g \times 778} \{1 - \sin^2 22^{\circ}\frac{1}{2} - (\cos 22^{\circ}\frac{1}{2} - 0.6)^2\} = 0.000015 V_1^2.$$

If the peripheral speed of the intermediate cylinder be taken as a mean proportional between those for the small and large cylinders, we shall have for it the speed of about 210 feet per second.

The heats per stage may be calculated as follows:

Cylinders.	Peripheral speed.	Steam velocity.	Squares.	Heat required.	Heat per stage.
1	2	3	4	5	6
Small . . . . .	135	225 ✓	50,625	0.759	2.025
Intermediate . . . . .	210	350 ✓	122,500	1.837	4.9
Large . . . . .	330	550 ✓	302,500	4.537	12.1

The steam velocities are first found by dividing by the ratio 0.6, and the squares are set down in column 4. Then the heat required for accelerating the steam to that velocity is found by equation (4), and set down in the fifth column. But the overall heat factor has been taken as 0.75, so that the heat to be allowed per row of blades or guides is found by dividing the quantities in column 5 by that ratio; again the heat per stage is twice the heat per row; so that the quantities in the sixth column are found by multiplying the fifth column by

$$2 \div .75 = 2.667.$$

If all the stages were on the small cylinder there would be

$$322.2 \div 2.025 = 159 +,$$

found by dividing the total adiabatic heat by the heat for a small cylinder stage. One-third of this gives 53 stages for the small cylinder. The stages for the intermediate and large cylinders are inversely proportional to the heats per stage, giving

$$\text{Intermediate, } \frac{53 \times 2.025}{4.9} = 22 -;$$

$$\text{Large, } \frac{53 \times 2.025}{12.1} = 9 -.$$

These numbers of stages are a general guide only because an exact distribution is neither possible nor important. We will assign 10 stages to the large cylinder, 20 to the intermediate cylinder, and will assign the remainder of the available heat to the small cylinder, and compute the number of stages as follows:

$$322.2 - 10 \times 12.1 - 20 \times 4.9 = 103.2;$$

$$103.2 \div 2.025 = 51. \checkmark$$

That is to say, the remainder of the available heat, after allowing for the intermediate and large cylinders, is 103.2 thermal units, and this corresponds to 51 stages of the small cylinder. Assuming that there are to be three barrels per cylinder, we may assign the number of rows per barrel at discretion, remembering that the blade heights are liable to increase rapidly for the lower stages, and therefore assign the larger number of rows to the earlier barrels.

Cylinders.	First barrel.	Second barrel.	Third barrel.
Small . . . . .	18	17	16
Intermediate . . . . .	8	6	6
Large . . . . .	4	3	3

All the blades of a barrel are given the same height, and it will be assumed that the best arrangement will be made if the blades

have the correct height at the middle. Following our general method, with necessary modifications, we will determine the temperatures and specific volumes at the middle of the barrels, and determine the proper blade lengths at those points.

There are 18 stages on the first barrel, so that 9 stages will come before the mid point and 9 after. Nine stages will call for

$$9 \times 2.025 = 18.23 \text{ B.T.U.}$$

for the first interval. The second barrel has 17 stages so that from the middle of the first stage to the middle of the second there will be

$$9 + 8\frac{1}{2} = 17\frac{1}{2} \text{ stages}$$

calling for

$$17\frac{1}{2} \times 2.025 = 35.44 \text{ B.T.U.}$$

for the second interval. Proceeding in this way the intervals can be found as follows:

$9 \times 2.025$	$= 18.23$
$(9 + 8\frac{1}{2}) 2.025$	$= 35.44$
$(8\frac{1}{2} + 8) 2.025$	$= 33.41$
$8 \times 2.025 + 4 \times 4.9$	$= 35.8$
$(4 + 3) 4.9$	$= 34.3$
$(3 + 3) 4.9$	$= 29.4$
$3 \times 4.9 + 2 \times 12.1$	$= 38.9$
$(2 + 1\frac{1}{2}) 12.1$	$= 42.35$
$3 \times 12.1$	$= 36.30$
$1\frac{1}{2} \times 12.1$	$= 18.15$
	<hr/>
	322.28

In further explanation it may be said that in passing from the middle of the third barrel to the middle of the fourth barrel there will be 8 stages at 2.025 B.T.U. per stage, and 4 stages at 4.9 B.T.U. per stage. Finally, there are  $1\frac{1}{2}$  stages on the ninth barrel from the middle to the end which are included in our calculation, because it gives a valuable check.

Cylinders.	Barrels.	Stages.	Mid stage.	Heat portions.	Factors for temperature distribution.	Heat assignments.	Heat contents entropy 1.56.	Temperatures.	Absolute pressures.	Heat contents for volumes.	Specific volumes.	
I	2	3	4	5	6	7	8	9	10	11	12	13
I	...	...	...	...	...	...	1193.3	366	164.7	1193.3	...	...
	1	18	9	18.23	1.047	19.1	1174.2	348	131.1	1179.6	3.38	1.60
	2	17	26.5	35.44	1.038	36.8	1137.4	314	82.3	1153.1	5.15	2.43
	3	16	43	33.41	1.028	34.3	1103.1	283	51.6	1128.0	7.85	3.71
II	4	8	55	35.80	1.017	36.4	1066.7	252	30.9	1101.2	12.47	2.44
	5	6	62	34.3	1.006	34.5	1032.2	223.5	18.4	1075.4	19.8	3.87
	6	6	68	29.4	0.997	29.3	1002.9	200	11.53	1053.4	30.5	5.96
III	7	4	73	38.9	0.987	38.4	964.5	170	5.99	1024.3	55.2	4.37
	8	3	76.5	42.35	0.974	41.2	923.3	139	2.81	992.5	109.9	8.70
	9	3	79.5	36.3	0.961	34.9	888.4	114	1.43	965.3	204.5	16.20
	...	...	...	18.15	0.953	17.3	871.1	102	1.00	...	...	...
				322.28								
						322.2						

From page 68, it appears that the overall heat factor 0.75 gives for the ratio for temperature distribution 1.05; Fig. 83 is

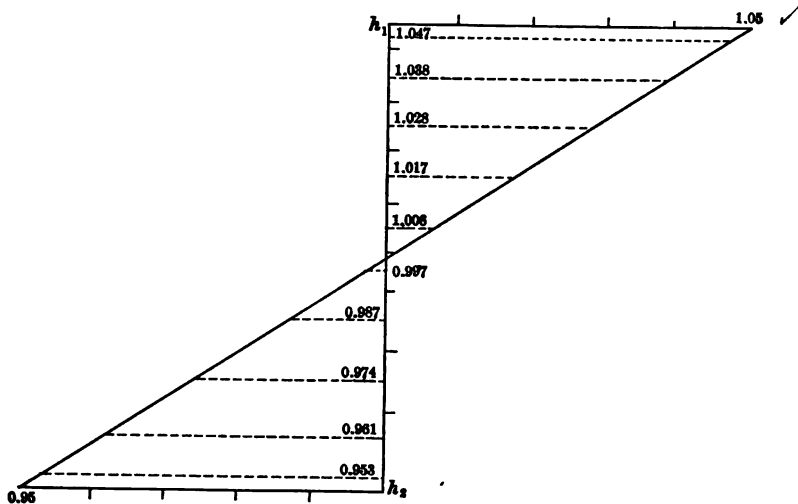


FIG. 83.

drawn with this ratio for finding the factors for temperature distribution, which are transferred from this diagram to column 6

of the table. In laying off the heat portions  $h_1$   $h_2$  of Fig. 83, it is convenient to make that line represent the total available heat 322.2 B.T.U., and then the heat portion of column 5 can be laid off in succession to the same scale.

The heat assignments of column 7 are the products of the quantities in column 5 by the factors in column 6. These heat assignments at entropy 1.56 are now subtracted in succession from the original heat contents (1193.3), giving the heat contents of column 8. Now we find in the entropy table the corresponding temperatures and pressures of columns 9 and 10. To find the specific volumes, we find the heat contents of column 11, by subtracting in succession the products of the heat portion of column 5 and the overall heat factor; thus

$$1193.3 - 0.75 \times 18.23 = 1179.6;$$

$$1179.6 - 0.75 \times 35.44 = 1153.1, \text{ etc.}$$

The specific volumes are found by interpolating in the entropy table at the proper temperatures, to correspond with the heat contents of column 11.

Before determining the blade lengths we must find the diameter by equation (5), page 206.

$$d_1 = \frac{60 \times 135}{750 \pi} = 3.44 \text{ feet} = 41.3 \text{ inches};$$

$$d_2 = \frac{60 \times 210}{750 \pi} = 5.35 \text{ feet} = 64.2 \text{ inches};$$

$$d_3 = \frac{60 \times 330}{750 \pi} = 8.4 \text{ feet} = 100.8 \text{ inches.}$$

The number of revolutions has been selected from the table on page 204, and given in the data; and the several peripheral speeds are taken from page 216.

The steam per shaft horse-power per hour was computed to be 11.65 pounds. Consequently the steam for 8500 horse-power is

$$\frac{11.65 \times 8500}{60 \times 60} = 27.5 \text{ pounds per second.}$$

The equation for blade length, with the standard ratio of thickness of blade to clear space (1 : 3), is found on page 207,

$$h = \frac{144 \times 4 w v}{3 \pi d V_1 \sin \alpha}$$

Using the steam velocities on page 216, we have for the three cylinders

$$\text{Small, } \frac{144 \times 4 \times 27.5}{3 \pi 41.3 \times 225 \times 0.383} = 0.4722.$$

$$\text{Intermediate, } \frac{144 \times 4 \times 27.5}{3 \pi 64.2 \times 350 \times 0.383} = 0.1953.$$

$$\text{Large, } \frac{144 \times 4 \times 27.5}{3 \pi \times 100.8 \times 550 \times 0.383} = 0.0792.$$

The specific volumes of the preceding table, multiplied by the proper one of these factors, give the blade lengths as set down.

All the blade lengths, except the last, come within the limits of Speakman's rule, which says that the length should not be less than 0.03 nor greater than 0.15 of the pitch diameter; the last length is 0.16 of the pitch diameter of the large cylinder, and is therefore slightly in excess.

The tip clearances, as computed by the equation on page 208, together with the ratio to the blade lengths, are given in the following table:

Cylinder.	I.			II.			III.		
Barrel . . .	I	2	3	4	5	6	7	8	9
Blade length	1.60	2.43	3.71	2.44	3.87	5.96	4.37	8.70	16.2
Clearance . .	0.037	0.037	0.037	0.053	0.053	0.053	0.077	0.077	0.077
Ratio . . .	0.023	0.015	0.010	0.022	0.014	0.009	0.018	0.009	0.005

Turning to the table on page 209, we may select the width, pitch, and axial clearance as follows:

Barrel . . . . .	1	2	3	4	5	6	7	8	9
Width . . . . .	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$2\frac{1}{8}$ "	$2\frac{1}{8}$ "
Pitch . . . . .	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$1\frac{1}{8}$ "	$2\frac{1}{8}$ "	$2\frac{1}{8}$ "
Axial clearance . . . . .	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "	$\frac{1}{16}$ "

To avoid using too large a variety of blading material the third barrel may be made like the first and second, and the sixth like the fifth and seventh.

Having the diameters of the pitch surfaces of the several cylinders, the blading can be laid out much as is indicated by Fig. 78, on page 209, except that for a stationary turbine the blades are set half inside and half outside the pitch surface, and consequently the diameters of the several barrels of a cylinder decrease as the blade lengths increase.

It will be noted that the blade lengths are counted as reaching from the surface of the rotor to the case; the actual lengths of the blades are, therefore, shorter than the computed lengths by the amount of the clearance. The guides are affected by the same consideration.

At the end of a barrel there is a double axial clearance allowed: first the normal clearance and second an additional distance equal to half the increase in blade height. When the blades are half inside and half outside the pitch surface, this extra clearance, marked  $\frac{A}{2}$  in Fig. 78, may be made equal to half the increase at one end, or one-fourth the increase in computed length.

**Discrepancy in Blade Lengths.** — When there are only a few barrels for a turbine, the blade lengths change at considerable intervals and there is a notable discrepancy between the assigned blade lengths at the beginning and end of a barrel, and the lengths appropriate to those places. This condition is exhibited by Fig. 84, which is drawn with heat portions per barrel as abscissæ and blade lengths in inches for ordinates. It will be seen that our distribution, which was to some extent arbitrary, is fairly even, the excess length at the beginning of a barrel being about the same proportion of the true length as the deficit at the end is of the proper length there. For the first two cylinders the discrepancy is somewhere near twenty per cent, but for the third cylinder, and especially for the last row of blades, it is larger; the condition for the last row is after all not

so bad, as it is customary and convenient to reduce the area of passages in the last stage of a turbine, to provide for conditions that are liable to be found for reduced powers.

There are three methods that can be used for remedying this condition. The most obvious method, if conditions of manufacture would admit, would be to increase the number of barrels; twice the number would about halve the discrepancy. Another method is to twist the blades after they are set, and so reduce the effective area for flow of steam at the beginning of the barrel,

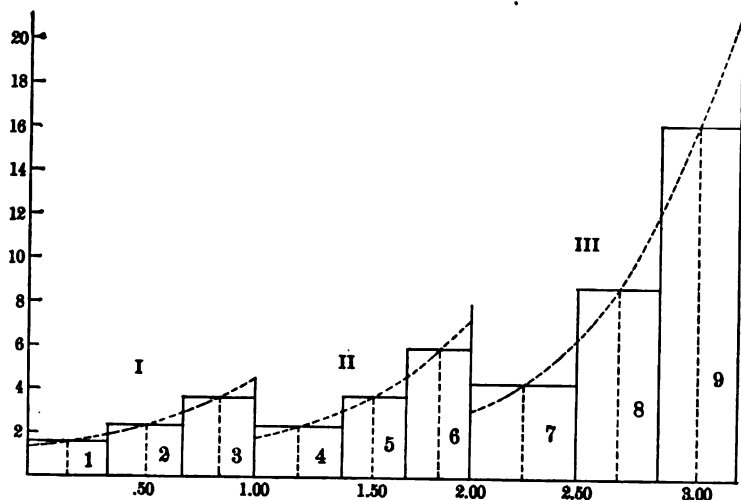


FIG. 84.

and increase it at the end. By this method, both the effective area and the steam velocity will be changed; the latter may be investigated by a diagram like Fig. 76; the area will vary nearly as the sine of the angle for small changes. The third method is to vary the number of blades; it is direct, but not very effective. Suppose, for example, that it were desired to increase the area by 33 per cent when the thickness of the metal is a third of the clear space; then all the blades would need to be removed.

In order to avoid excessive blade lengths at the low-pressure end of a marine steam-turbine, the exit angle is increased and the

number of blades is reduced. If the normal angle of exit is  $20^{\circ}$  to  $25^{\circ}$ , the last rows may have that angle increased to  $30^{\circ}$  or  $45^{\circ}$ . This at once increases the steam velocity and the effective area. If the number of blades were maintained the same as for earlier stages, there would be a considerable reduction of the ratio of thickness of metal to clear space, and the blades would appear to be spaced wider apart. Removing some of the blades increases this appearance of sparse blading, even if it does not much increase the effective area.

**Marine Turbines.**—The earliest steam-turbines applied to marine propulsion were of the Parsons type, and that type has now been so developed that it is used with certainty. Sections of marine turbines will be found opposite page 274. Parsons marine turbines are usually applied with three or four propellers with their shafts. When three shafts are used there is one high-pressure turbine on the middle shaft, and two low-pressure turbines, one on each of the wing shafts. When there are four shafts there are two high-pressure and two low-pressure turbines. The distribution of work among the several turbines will be discussed in Chapter X; the general problem of marine propulsion will be treated in Chapter XI.

## CHAPTER IX

### ACCESSORIES

THERE are certain accessories that are or may be applied to various types of steam-turbines, and which it is convenient to treat collectively, even though illustrations are drawn from certain special makes of turbines.

**Governors.** — There are two ways of controlling the power developed by steam-engines: (1) by varying the steam-pressure, and (2) by regulating the volume of steam supplied. For reciprocating engines we have throttling governors and cut-off or automatic governors. Throttling governors can, of course, be applied to any fluid-pressure motor, and are well adapted for all types of steam-turbines; some steam-turbines, like the Parsons turbine, can be controlled in no other way. Impulse turbines, with a relatively small number of nozzles for the first stage, may be controlled by opening more or less of them. Marine engines, including marine steam-turbines, may be controlled by hand, by varying the opening of the throttle valve, or by opening the proper number of nozzles of an impulse turbine. Stationary steam-turbines are habitually controlled by some form of revolving governor, and such governors are applied also to the Parsons marine turbine.

There appears to be no reason why a sensitive and powerful revolving governor cannot be made to control any steam-turbine with sufficient closeness and certainty. This method is applied by the Willans Company to turbines of the Parsons type.

**De Laval Governor.** — This simple type of steam-turbine, which runs at a very high number of revolutions, has a compact and powerful governor of the revolving type, in which the centrifugal force is balanced by a spring. The idea of the governor

is shown by Fig. 85. Here  $S$  is the main shaft of the turbine which carries the revolving weights  $bb'$ . The bent levers  $bef$

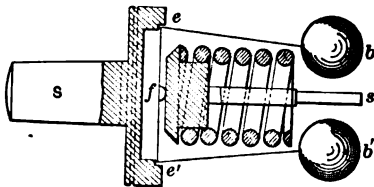


FIG. 85.

and  $b'e'f$  tend to force the spindle  $s$  out against the helical spring, and so operate on the throttle valve. The general arrangement is shown by Fig. 86 where again  $S$  is the main shaft of the turbine and  $s$  is

a spindle which acts on the lever  $E$ . The revolving weights are two half cylinders  $bb'$  with knife-edges at  $e$  and  $e'$  and

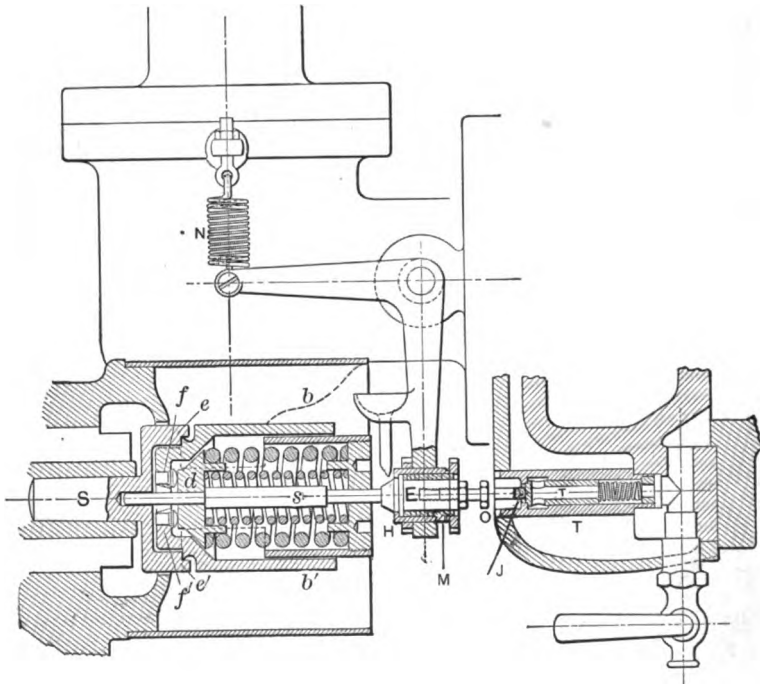


FIG. 86.

with projections  $ef$  and  $e'f'$ , forming bent levers, which bear against  $d$ . When the speed increases, the springs are compressed a little further and the spindle  $s$  is thrust out against the lever  $E$

and partially closes a throttle valve in the steam pipe. The shaft, on which the lever arm *E* is fixed, extends into the steam pipe, as shown by Fig. 87, and operates a double-beat valve at the end of a horizontal arm. Returning to Fig. 86, it is seen that there is an external spring at *N* which adds to the load on the governor. The safety device consists of a vacuum breaker at the right of the governor. When the device is in action the pet cock is open, so that air may be admitted to break the vacuum by opening the valve on the hollow spindle *T*. At *M* there is a stiff spring that keeps the piece *H* hard up against its shoulder, and this piece and the arm *E* work as one piece, so long as the governor has control of the throttle valve. If the turbine still continues to run overspeed with the throttle valve down hard on its seat, the spring at *M* yields, and the end *O* of the spindle comes in contact with *J* and opens the valve at *T*, thus admitting air and spoiling the vacuum. This governor is described at some length, because it is simple, powerful, and direct, and has an important feature of selecting first the ordinary throttle valve and afterwards the emergency safety valve or vacuum breaker.

**Parsons Governor.** — This governor was invented by Mr. C. A. Parsons and is very widely used for his type of turbine for marine as well as stationary work. The most notable feature is a jiggling device which gives a governing by "gusts". It is also a relay governor, but the two features have no necessary relation to each other. The governor is commonly geared to the turbine shaft by helical gearing, to run at a convenient number of revolutions, usually less than the turbine.

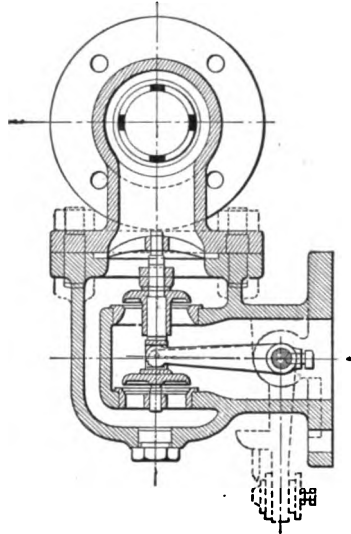


FIG. 87.

In Fig. 88, *aa* is the governor spindle carrying the revolving balls *b* and *b'* which act through the bent levers *bef* and *b'e'f'* on the spring plate *R*; above this plate is a helical spring not shown in the figure. The spindle is broken off above the plate; it should extend above *R* a distance greater than *ee'*, and have a device for adjusting the spring tension and, therefore, the speed of the turbine.

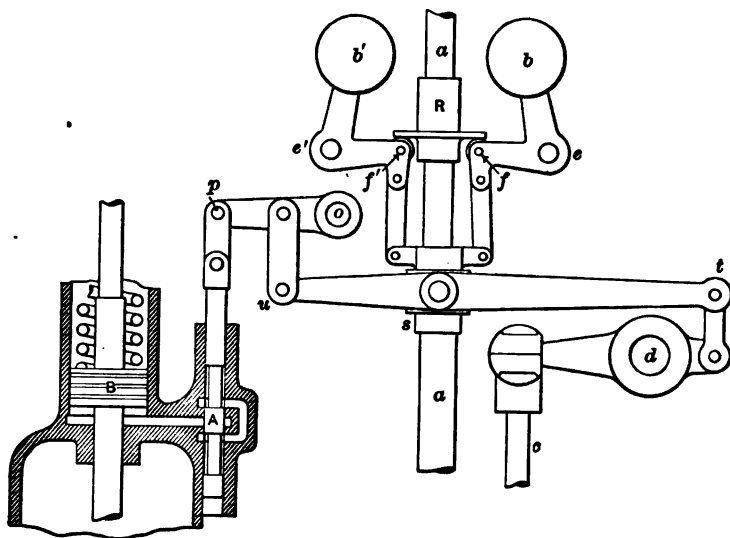


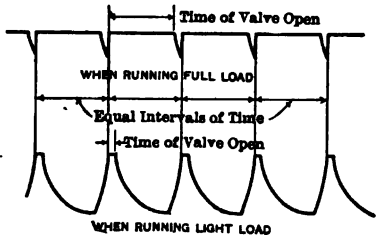
FIG. 88.

Instead of acting directly on the throttle valve, the governor controls the relay valve *A*, which controls the steam supply to the piston *B*, and this piston moves the throttle valve. There are two advantages from this relay: in the first place, the governor has a light duty and need not be made so large and powerful as would be necessary for direct action, and, in the second place, the turbine may run at nearly the same speed at all powers, because when the piston *B* has moved far enough to give the proper steam supply, the valve *A* is returned to mid position by the governor. This last feature is found in all properly-

designed relay governors, and has long been used for water-wheel governors.

The jiggling device is *cdt*, which receives an oscillation from an eccentric on the shaft that drives the governor. Consequently the pivot *t* of the lever *tsu* has a small reciprocation, which continually opens and closes the valve *A*, and consequently forces the piston *B* to continually open and close the throttle valve. Since the governor is free to yield to any force applied to it, the sleeve *s* has also a slight reciprocation, which keeps the governor moving and makes it a little more sensitive. Many governors for reciprocating engines have a little jiggling motion transmitted to them from the valve gear, either intentionally or accidentally, which some engineers consider advantageous.

When the turbine is under full load the throttle valve is closed only a very brief portion of a reciprocation of the gear; when the load is light the valve opens only a small part of a reciprocation. This action is represented by Fig. 89, and is said to govern by gusts. When the turbine is under full load there is nearly a continuous admission at full pressure; at light loads there are brief intervals at full pressure. There appears to be no advantage from this action, as there are from 100 to 400 gusts a minute depending on the period of the jiggling, and by the time the steam gets to the turbine blades the pressure must be nearly uniform.



**Governors for Impulse Turbines.** — As already pointed out, and as illustrated for the de Laval turbine, an impulse turbine can be controlled by a throttle valve. The de Laval turbine can also be regulated in part by hand, opening or closing valves that admit steam to the nozzles. But the preferable method is to admit steam to a portion of the set of nozzles when running at less than full load. Fig. 90 shows how this can be done, simply and directly, by a slide valve that may be moved

by the governor to shut off steam from some of the nozzles. One nozzle is likely to work under unfavorable conditions, especially when open but a little.

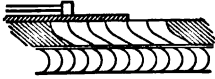


FIG. 90.

The Rateau turbine is said to be controlled by a combination of such an obturator and a throttle governor; the obturator, or slide, may be moved by a slow and positive mechanism, the balance of effort of the turbine to the work demanded being maintained by the throttle governor. According to the general practice, overloads are met by introducing steam to one of the lower stages of the turbine. There are mechanical difficulties in the way of sensitive control of a turbine by such a slide or obturator, especially as lubrication by oil is prohibited.

It has sometimes been proposed to vary the number of nozzles in action in the several stages of a pressure-compound turbine, so as to maintain the proper pressure in the stages, and the normal relation of the steam and peripheral velocities, but, though attempts have been made, the practice has not received favor. If the first set of nozzles only is controlled, the action of the first stage will be normal when it has a large drop of pressure; but all the other stages will have reduced pressures and less steam velocities. If the turbine is designed for full power, and with the nozzle areas that the direct method of design would assign for the last stage, that stage is liable to run idle; to avoid such a contingency the last set of nozzles may be given less area than such a design would indicate; they will consequently have a greater pressure drop and greater steam velocity than the other sets of nozzles when running under power.

**Rice's Valve Gear.** — The Curtis turbines, as built by the General Electric Company, are controlled by a gear invented by Mr. R. H. Rice, and illustrated by Figs. 91 and 92. The governor, of the spring-loaded revolving type, is directly connected to the main shaft of the turbine, and is consequently compact and powerful. It acts as a relay governor, so that its duty is light.

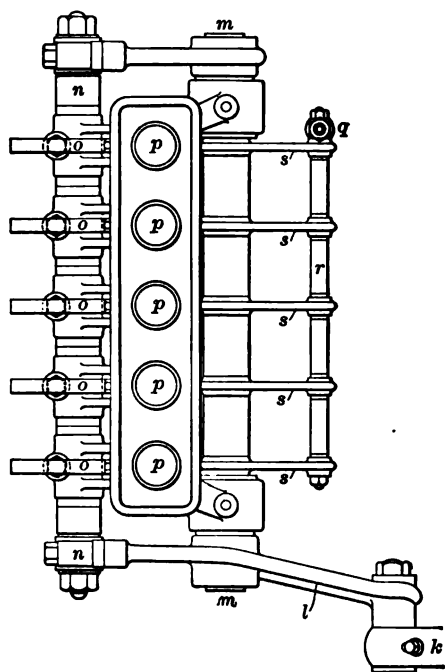


FIG. 91.

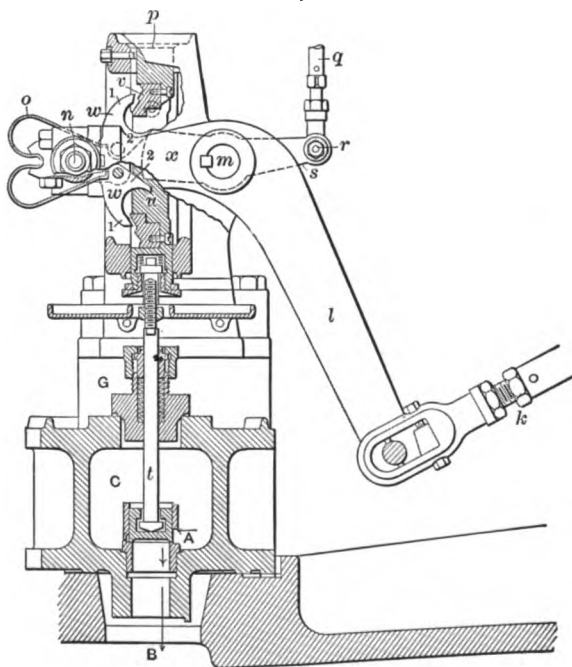


FIG. 92.

The first stage of the turbine has the nozzles in one large group, or for very large powers, in two groups. A large group is divided into small groups, usually three nozzles to a group, each of which is supplied with steam by a poppet valve; Fig. 91 shows five such groups.

The valve shaft *mm* carries the mechanisms for moving the valves and also the governing devices. On this shaft is a rocking frame *mmnml*, which has a comparatively slow reciprocation given to it through the lever *l* and the connecting rod *k*, from an eccentric which is geared to the turbine shaft. Opposite each valve spindle there is a piece which carries two dogs *w, w*, one pointing up to open the valve, and one down to shut the valve; they may engage in notches on the valve-spindle head *uv*, either at the top or at the bottom.

The governor is attached to the controlling gear by the rod *q*; this gear consists of a set of cams *sx*, one for each valve, which may trip the dogs so that they cannot catch the notches in the valve-spindle heads. In Fig. 92 the upper dog is tripped by the cam and cannot open the valve, but the lower dog is shown in position to shut the valve. There is a spring under the valve-spindle head to give a little elasticity. The governor has therefore only the light duty of placing the cams, the valves being moved positively by the turbine.

The several cams are set at a slightly increasing angle, so that under normal conditions the valves at one end of the large group are shut, and at the other end are open; a valve in the middle is opened and shut for each reciprocation of the rocking frame, the period of opening being adjusted to meet the demand for power. If the demand for power increases, that valve stays open and the next valve is opened and shut, etc. It is to be noted that all the nozzles but those of one group are either open or shut, and that one group is alternately open and shut, and therefore they all act under normal conditions without throttling.

It is further arranged that when all the nozzles are open, and further demand for power is to be met, supplementary over-

power nozzles are opened to supply steam to the blades of one of the lower stages.

**Hydraulic Valve Gear.** — For large powers the General Electric Company use the hydraulic mechanism, shown by Fig. 93. Here *A* is a piston that is operated by oil pressure and acts directly on the valve mechanism; it is controlled by the valve *V* through the floating lever *def*, which is connected to the

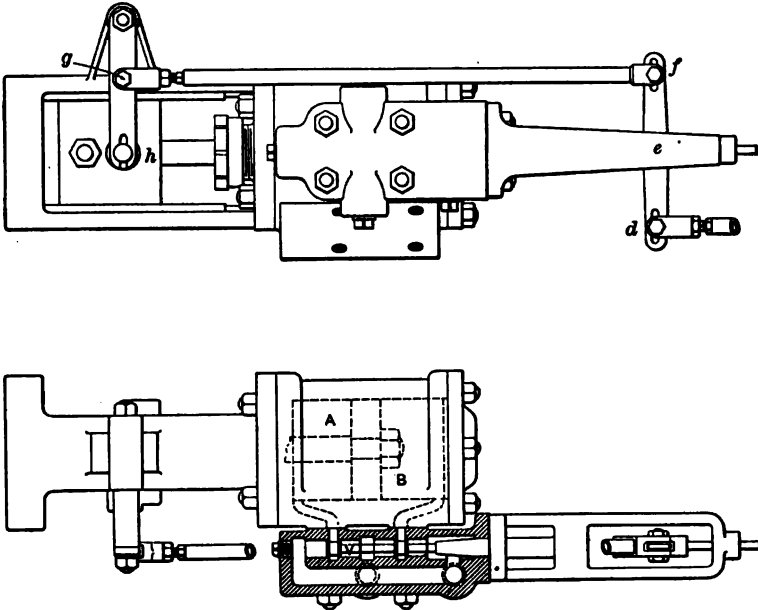


FIG. 93.

governor at *d*, to the valve spindle at *e*, and to the cross head at *f*; the connection to the cross head is by the linkage *hgf*. Suppose that the valve *V* taking oil in the middle is moved to the right by the governor; the oil flows in at the right end of the cylinder and out at the left, and the piston *A* moves to the left. But the cross head now moves the point *f* of the floating lever to the left, and so returns the valve *V* to mid position, ready for the next signal from the governor. The hydraulic gear can be

made powerful enough to act under all conditions, whether the gear is properly lubricated and in good condition or not.

The valves leading to groups of nozzles are arranged much as shown by Figs. 91 and 92, but the gear is powerful enough to operate the valves by cams directly. These cams are all on one cam shaft but set at varying angles, so that the operation of the valves is the same as by the mechanical gear previously described.

**Oil System.** — All the bearings of a steam-turbine must be flooded with oil under pressure, and the oil must be cooled, circulated, and strained continuously, for the slightest defect in lubrication is likely to give trouble immediately. Fig. 94 shows the oil system of the vertical turbines built by the General Electric Company. There is a large storage tank with a device for cooling the oil not shown in the figure. From this tank the oil is forced by a special pump to all the bearings of the turbine and generator; from those bearings the oil flows back to the tank. To equalize the flow there is, on the main oil line, either an accumulator with ram and weight, or a spring accumulator which acts much in the same way. There is also an air chamber to mitigate pulsations from the pump. The vertical turbine has a footstep bearing which requires high pressure of oil; advantage is taken of this high-pressure oil to operate the hydraulic valve gear. The other bearings, which do not need high pressure, are supplied through a reducing valve.

**Shaft Glands.** — The Curtis turbine, as built by the General Electric Company or by the Fore River Shipbuilding Company, has shaft glands provided with carbon rings, which are an inch square or more in section. They are made in three or more segments, halved together at the ends, so that when assembled they form a complete ring. The rings are turned out inside a trifle larger than the shaft, and when assembled in the gland are expected to nearly, but not quite, touch the shaft; should they touch they wear away without injuring the shaft. At the high-pressure end where the steam leaks into the atmosphere there may be three or four sets of such rings, put in singly or in pairs; at the low-pressure end where air tends to leak into the turbine

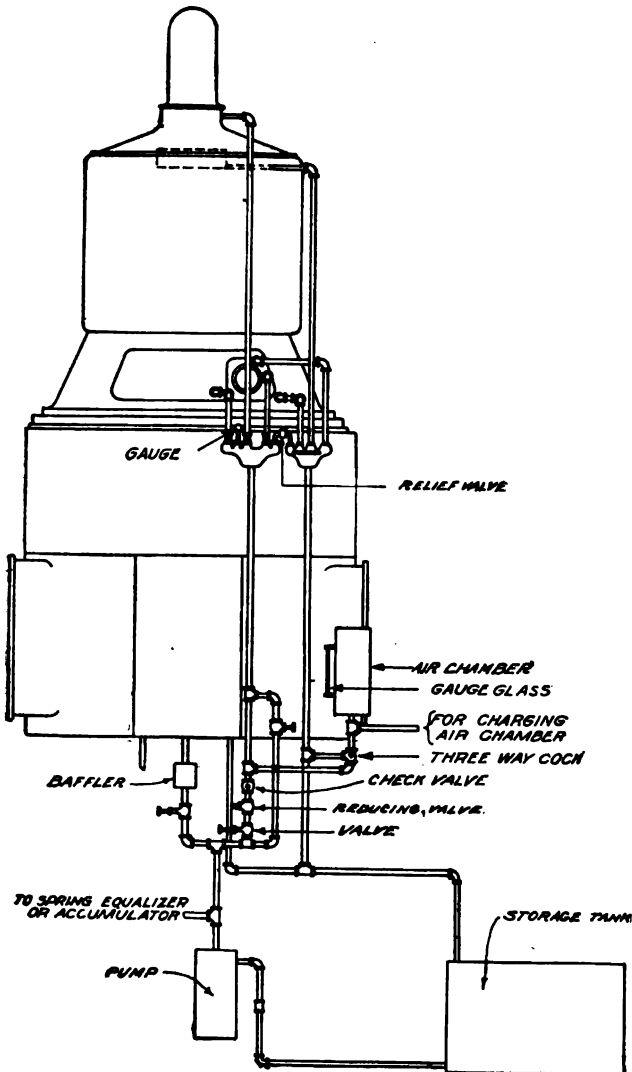


FIG. 94.

be two or three sets, with steam supplied between, so the leakage shall be of steam into the turbine or against the atmosphere.

**Westinghouse Water-seal.** — The leakage past the shaft of the Rateau type of turbine, illustrated on page 120, is prevented by a special water-seal made by a revolving paddle wheel about 28 inches in diameter. The diameter of the pitch surface is 7.25 feet, consequently if the peripheral velocity be taken as 300 feet per second, the velocity at the edge of the paddle wheel will be  $7.25 \times 12 : 28 :: 300 : 96.6$  feet per second. The centrifugal acceleration corresponding will be

$$\frac{v^2}{r} = \frac{96.6^2}{14 \div 12} = 7990,$$

where  $v$  is the velocity of the edge of the paddle wheel in feet per second, and  $r$  is the radius in feet. This is

$$7990 \div 32.16 = 250 -$$

times the acceleration due to gravity. Now the weight of a cubic inch of water is about 0.036 of a pound, consequently the pressure of one inch of water with an acceleration 250 times that of gravity will be

$$250 \times 0.036 = 9 \text{ pounds,}$$

therefore a water-seal of an inch and three-fourths of water will be sufficient to exclude the atmosphere. That is, the radial width of the rotating band of water must be an inch and three-quarters wider inside than outside.

**Labyrinth Packing.** — An important feature to prevent excessive leakage past the dummy or balance cylinders is the labyrinth packing, illustrated by Fig. 95, which is taken from Speakman's paper. Here the rotor is cut with a series of grooves into which brass strips project from the casing. The rotor is adjusted so that the axial clearance between the strips and the edge of the groove is from 0.007 to 0.015 for electrical work, and somewhat more for marine work. The steam in passing through the packing is alternately wire-drawn and checked, so that the

flow is much reduced. Sometimes such a packing is made by setting strips of brass in both rotor and casing. For marine work the expansion of the cylinder on account of steam temperature

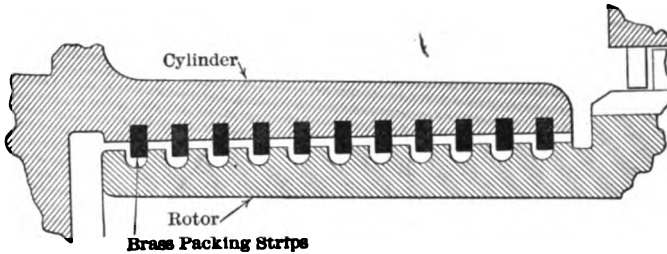


FIG. 95.

is so much that a labyrinth packing remote from the thrust bearing cannot be adjusted to give small axial clearance; in such case the packing consists of strips set in both rotor and casing, which are given as small a radial clearance as possible. The strips are thinned at the edge, as shown by Fig. 96. The proper radial clearance may be computed by the equation on page 208; since there is no risk of stripping, the clearance may be somewhat less than the rule indicates.

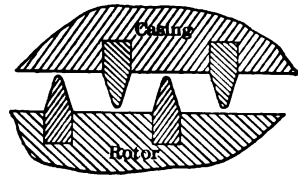


FIG. 96.

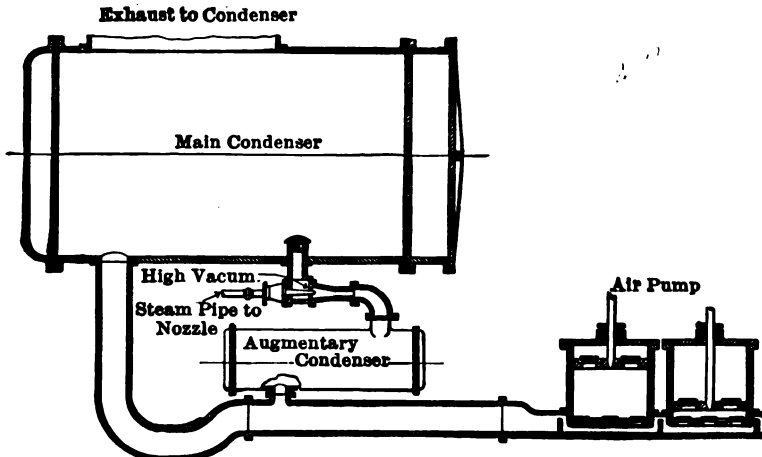


FIG. 97.

The labyrinth packing has been used also as a shaft gland, either by itself or in conjunction with some other device, as for example, with the Westinghouse water-seal.

The diaphragms between pressure stages of impulse turbines are frequently grooved, and so become analogous to the labyrinth packing; this is illustrated by Fig. 41, page 118.

**Vacuum Augmenter.** — Great importance is attributed to obtaining a good vacuum for steam turbines, and to this end

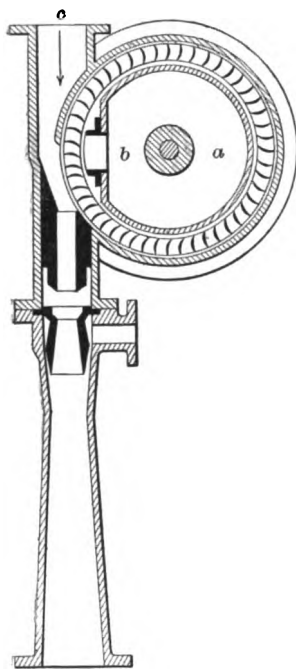


FIG. 98.

Mr. Parsons has invented a vacuum augmentor, shown by Fig. 97, as applied to the S. S. *Victorian*. It consists essentially of an ejector which draws air and vapor from the condenser, and delivers them through an augmentary condenser to the air pump. It is claimed that a vacuum of  $27\frac{1}{2}$  to 28 inches of mercury can be realized in the condenser, while the air pump is required to develop only 26 inches. The air pump must be set sufficiently below the condenser to get a head of water to balance the difference in vacuum. The ejector is estimated to use one per cent as much steam as the turbine, and to produce an appreciable gain in economy, measured as usual in pounds of water per horse-power per hour.

**Wet and Dry Air Pumps.** — It is customary to use both wet and dry air pumps when a good vacuum is desired, both for reciprocating engines and turbines; as the importance of a high vacuum is insisted upon by turbine builders, they more frequently provide this arrangement. The dry air pump draws from that part of the condenser which is likely to show the least proportion of moisture in the air; the wet air pump is usually of the ordinary type of air pump, but its

office is to remove the water. Sometimes a compound centrifugal pump is used for the wet air pump.

**Le Blanc Air Pump.**—The Le Blanc dry-air pump is shown by Fig. 98, as made by the Westinghouse Company. It may be described as an aspirator and ejector which draws air and vapor from the air space of a condenser. At *a* there is a chamber which is supplied with water at low temperature, about  $70^{\circ}$  F., from which the water flows through the nozzle at *b*. Outside the chamber a wheel with a large number of blades revolves with high velocity. At *c* air comes in from the condenser. The wheel draws water from the orifice *b* and throws it in broken masses through the air into the ejector; in consequence the air is drawn down from *c*, mingles with the water, and the combined air and water are driven out through the ejector nozzles.

## CHAPTER X

### EFFECT OF CONDITIONS

IN the previous chapters computations and designs have been made in the simplest and most direct method; in some cases the effect of certain conditions of construction or service which applied to the type in question was afterwards considered. But there are conditions that affect all types of turbines or, in fact, all steam engines. The effect of some such conditions will be treated in this chapter; the variety of conditions and combinations that may come up in practice is too great to be treated exhaustively, but our end will be attained if the student learns how to approach the problems that arise.

**Efficiency and Economy.** — In order to apprehend the problems that may be brought up by conditions of service of steam turbines, the student should be familiar with the general theory of thermodynamics and its application to steam-engines, as presented in any good textbook. In the author's textbook,\* Chapters VIII, IX, and XII will be found pertinent to this object.

The thermal efficiency, either for the ideal Rankine cycle or for the actual engine, as treated on pages 33 and 35 of this book, is sufficiently clear and definite. Again, the relation of those efficiencies is shown by the overall heat factor, so that the conception of the relative efficiency of steam engines, and in particular of steam turbines, should be well in mind.

There is, however, one phase of the question which has been treated in the conventional way without comment, that is to say, the statement of economy in terms of the consumption of steam in pounds per horse-power per hour. The variant, steam or water rate per kilowatt, involves only a factor to pass from one unit of energy to another.

\* Thermodynamics of the Steam Engine.

Now, from the construction of turbines, all, or nearly all, of the steam enters the turbine directly and passes entirely through. There is nothing akin to the use of steam-jackets for reciprocating engines. In consequence, our treatment of economy is entirely consistent, so long as the conditions of the actual turbine and those assigned to the ideal cycle are the same, including initial pressure, vacuum, and condition as to priming or superheating. But as soon as we make comparison of one turbine with another, or of one set of conditions for a given turbine with a different set of conditions, we are liable to fall into confusion and error if the steam per horse-power per hour is the basis of comparison. More particularly is this so in the comparison of the economy of steam-turbines with the economy of high-class reciprocating engines which habitually are provided with steam-jackets.

In order to avoid any chance of confusion, comparisons of economy should be based on an absolute unit like the thermal unit, in the customary form of British thermal units per horse-power per minute. This method of stating economy is applicable to all forms of heat engines, including, of course, steam-engines and steam-turbines.

As deduced on page 37 the thermal units per horse-power per minute, for both moist and superheated steam, may be computed by the form

$$(C_1 - q_2) W \div 60, \dots \dots \dots (1)$$

in which  $C_1$  is the initial heat contents of the steam from the entropy table, and  $W$  is the number of pounds of steam used by the engine. The same form applies also to Rankine's cycle,  $W_R$  being the theoretical steam consumption.

When the steam consumption has been determined, either by experiment or by computation, the thermal units per horse-power per minute will be computed by the above form, but if we have the thermal efficiency we may compute by dividing the number 42.42 by that quantity, as already shown on page 37.

It is further to be remembered that the ratio of thermal units

per horse-power per minute for two engines or two conditions is always inversely proportional to the thermal efficiencies; this is not always true of the steam consumptions. These circumstances will be illustrated by considering the advantages of superheated steam and high vacua.

**Superheated Steam.** — Much efficiency is attributed to superheating for steam-turbines, it being asserted that with superheated steam there is less erosion of blades and smaller rotation losses.

To see the advantage from superheating, compare two tests on a certain turbine running at nearly the same load and with the conditions following:

Press. Abs. initial.	Press. Abs. final.	Superheat.	Steam per shaft horse- power per hour.	Approximate heat contents.	Heat of liquid.
187.8 188.7	1.01 1.02	48.4 145.5	11.8 11.0	1231 1278	70 70

These conditions are found approximately at the temperature  $377^{\circ}$  and at the entropies 1.59 and 1.64, at which the heat contents are as given above; the heat of the liquid is nearly 70. The thermal units per horse-power per minute are

$$(1231 - 70) 11.8 \div 60 = 228,$$

$$(1278 - 70) 11.0 \div 60 = 222.$$

The ratio of the steam consumptions is

$$11.8 \div 11.0 = 1.07,$$

but the ratio of the thermal units is

$$228 \div 222 = 1.03,$$

and this is the true indication of the advantage of superheating  $145^{\circ}$  instead of  $48^{\circ}$ .

**High Vacuum.** — The steam-turbine is well adapted to take advantage of a good vacuum, and builders are careful to provide for this condition. Taking again results from tests on a turbine, we have the conditions:

Press. Abs. initial.	Press. Abs. final.	Superheat.	Steam per horse-power per hour.	Approximate heat contents.	Heat of liquid.
188.4	1.04	87.2	11.5	1249	71
188.7	2.4	111.3	13.35	1252	101

These conditions are found approximately at  $377^{\circ}$  and entropies 1.61 and 1.62; the heat contents are as given above, and also the heats of the liquid. The thermal units per horse-power per minute are

$$(1249 - 71) 11.5 \div 60 = 226,$$

$$(1252 - 101) 13.35 \div 60 = 256.$$

The ratio of the steam consumptions is

$$13.35 \div 11.5 = 1.16,$$

and the ratio of the thermal units is

$$256 \div 227 = 1.13.$$

The poorer vacuum has the advantage of  $24^{\circ}$  greater superheat, but from the preceding comparison this cannot be very important. The apparent advantage is not much greater in this case than the real advantage; but whatever be taken as the basis of comparison, the gain from the better vacuum is partially offset by the work of the air pump to maintain it. The absolute final pressures quoted correspond to 28 and 25 inches of vacuum respectively.

**Comparison with Reciprocating Engines.** — The assumed water rate of 14.5 pounds per kilowatt hour, for the turbine design on page 171, gives an estimated steam consumption per turbine horse-power of 9.72 pounds. The initial heat contents, on page 172, is 1271 and the final heat of the liquid 70; so that the heat consumption per horse-power per minute appears to be

$$(1271 - 70) 9.72 \div 60 = 195 \text{ B.T.U.}$$

This is a very good result as can be seen by comparing with the following performance of selected engines.

	Revolutions per minute.	Gauge pressure.	Horse-power.	Steam per horse-power per hour.	B.T.U. per horse-power per minute.
<i>Triple expansion engines</i>					
Leavitt pumping . . . . .	50.6	176	576	11.2	204
Sulzer mill engine . . . . .	56	149	1823	11.3	...
<i>Compound engines,</i>					
Horizontal mill engine . . . . .					
Superheated . . . . .	128	135	115	9.6	199
Saturated . . . . .	127	135	127	11.8	213
Leavitt pumping . . . . .	18.6	137	643	12.2	222

The most notable feature in the above table is the close concordance between the turbine in question and the compound engine using superheated steam. That engine, which developed only 115 horse-power, had the advantage of a large degree of superheat (256° F.), but on the other hand the steam pressure was only 135 pounds by the gauge. It is but fair to say that there would be little, if any, advantage from raising the steam pressure for that engine.

Comparing the performance of the turbine in question with that of the triple-expansion Leavitt engine, we have from the steam consumptions

$$11.2 \div 9.72 = 1.15;$$

but from thermal units

$$204 \div 195 = 1.05.$$

This comparison emphasizes the statement that comparisons of steam consumptions are liable to be misleading. In this case the discrepancy is due to two causes: first, the superheating for the turbine, and, secondly, the use of steam-jackets and intermediate reheaters.

**Equivalent Dry Steam.**—To allow for the advantage of superheating some engineers reduce the steam consumption to equivalent dry, saturated steam.

This process may be explained by an illustration. The turbine, for which a design is given on page 171, was estimated to

use 14.5 pounds per kilowatt hour, or 9.72 pounds per *turbine* horse-power.

The steam pressure was 180 pounds by the gauge, the vacuum 28 inches of mercury, and the superheating 125°. The initial condition is nearly the same as found in the entropy table at 380° and entropy 1.63, where the heat contents are 1271.4. The vacuum corresponds to 102° F. at which the heat of the liquid is 70. Consequently the heat required to raise a pound of water from 102° to 195.5 pounds, and to superheat it 129° is

$$1271.4 - 70 = 1201.4 \text{ B.T.U.}$$

By aid of Table II of the tables, we find that the heat required to raise a pound of steam from 102° to 195.5 pounds and to evaporate it into dry, saturated steam is

$$r_1 + q_1 - q_2 = 845.1 + 352.3 - 70 = 1127.4 \text{ B.T.U.}$$

The equivalent dry, saturated steam consumption is computed by multiplying the actual consumption by the factor

$$1201.4 \div 1127.4 = 1.065.$$

This gives

$$\begin{aligned} \text{Water rate per K.W. hour,} & \quad 14.5 \times 1.065 = 15.45; \\ \text{Steam per turbine H.P. per hour,} & \quad 9.72 \times 1.065 = 10.35. \end{aligned}$$

This method, though not precise, is not open to serious criticism when applied to steam-turbines, or to steam-engines without steam-jackets or reheaters. If we apply this result to compare the turbine with the triple-expansion Leavitt engine, on page 245, we get

$$11.2 \div 10.35 = 1.08$$

instead of 1.05, as found by comparing thermal units per horse-power per hour.

**Variation of Stage Pressures.** — As already said, the computations for design have been made in the most direct way, assuming normal conditions for full power throughout, although attention has been called to the fact that the last stage may develop little, if any, power when the turbine is running at reduced power,

and that the practical design may be modified to meet this condition.

For a rough preliminary investigation of changes of stage pressures for any cause, we may use Rankine's equations, on page 18, though these equations are properly restricted to calculation of flow of steam initially dry and saturated. These equations are

$$p_1 > \frac{5}{3} p_a, \quad w = a \frac{p_1}{70} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$p_1 < \frac{5}{3} p_a, \quad w = 0.029 a [p_a (p_1 - p_a)]^{\frac{1}{2}} \quad . \quad . \quad . \quad (2)$$

Equation (1) is applicable to impulse turbines with few pressure stages (six or less), which have throats for all the sets of nozzles, provided that the pressure changes do not change the relation of  $p_1$  to  $p_a$  from the first condition to the second. To apply to a given stage, we take  $p_1$  as the pressure in that stage, and  $p_a$  is the pressure in the next stage. Equation (2) may be applied when there are numerous stages, so that the nozzles have no throats.

It is apparent that from equation (1) we may estimate the stage pressures to vary directly with the weight of steam passing through the nozzles, and this conclusion is verified by experiments on turbines of this type, although the measurement of stage pressure and temperature is uncertain on account of the high velocities of steam in the stages.

In order to use equation (2) we must assume that the ratio of  $p_a$  to  $p_1$  is the same when the rate of flow of steam is changed as under normal conditions. If that assumption is justified, then this equation also indicates that the pressure  $p_1$  will vary directly with the weight of steam passing through the nozzles. We cannot expect so much satisfaction from the use of this equation as from the former.

Usually the estimated change will be sufficient for the purpose of the designer. Should he desire greater precision of calculation, he may use this method to get preliminary results,

and may then check the results by a step-by-step calculation like that on page 69. That is to say, the designer may compute the quality of the steam approaching the nozzle in question, by (1) subtracting from the initial heat contents, the equivalent of the work done per pound of steam to that point, and (2) finding by searching in the tables at the newly estimated stage pressure and the corresponding temperature, that entropy column which gives most nearly the heat contents remaining in the steam. He may then compute in that column the velocity and specific volume at the throat of the nozzle, and from them the weight of steam discharged. Should he not be satisfied with the result he may reestimate the pressure at discretion and try again.

To illustrate this, let it be assumed that the turbine to which the table on page 175 applies uses 16 pounds of water per kilowatt hour when developing 1500 kilowatts, instead of the 14.5 pounds at full load of 3000 kilowatts. The steam used in a unit of time with reduced load will be

$$\frac{16 \times 1500}{14.5 \times 3000} = 0.55$$

of what it is at full load. Neglecting changes in mechanical and electrical efficiencies we may estimate that the overall heat factor will change from that given on page 173 to

$$0.739 \times 14.5 \div 16 = 0.67.$$

The heat changed into work per pound of steam in each stage will be

$$72.2 \times 0.67 = 48.4 \text{ B.T.U.,}$$

instead of 53.4, as calculated for the normal condition.

Let us compute for the third set of nozzles which, by the table on page 177, have a total throat area of 25.1 square inches. The normal pressures from the table on page 175 are 33.6 and 19.5 at the entrance and throat; our method indicates

Initial pressure,  $33.6 \times 0.55 = 18.5$  pounds;

Throat pressure,  $19.5 \times 0.55 = 10.7$  pounds.

The heat contents of the steam approaching this third set of nozzles will be

$$1271.4 - 2 \times 48.4 = 1174.6,$$

and this figure comes nearly at  $224^{\circ}$ , and entropy 1.77; computing in this column we have

Entropy.	Approximate pressure.	Temperature.	Heat contents.
1.77	18.6	224	1177.3
1.77	10.6	196	1135.4
			<hr/> 41.9

Taking the friction factor as  $\gamma = 0.05$ , the heat for computing velocity becomes

$$41.9 \times 0.95 = 39.8,$$

so that the velocity is

$$223.7 \sqrt{39.8} = 1412 \text{ feet per second.}$$

The heat contents for finding volume is

$$1174.6 - 39.8 = 1134.8 \text{ B.T.U.,}$$

which at  $196^{\circ}$  gives 36 cubic feet. The weight of steam discharged per second is, therefore,

$$\frac{25.1 \times 1412}{144 \times 36} = 6.8 \text{ pounds of steam per second,}$$

instead of

$$12.08 \times 0.55 = 6.7$$

as would be estimated by our method.

A fair concordance may be expected for this method when the steam is nearly dry at the place chosen for calculation. If the steam is very wet or much superheated, the discrepancy may be large.

**Augmentation of Penultimate-Stage Pressure.** — To avoid running the last stage of a pressure-compound turbine idle and therefore as a brake, it may be advisable to arbitrarily increase the pressure in the preceding stage and to reduce the nozzle area

in the last stage correspondingly. If the approximate method of estimating the reduction of pressure, just given, is applied to the fourth stage of the turbine to which the table on page 175 belongs, then the pressure in that stage at half power would be

$$3.67 \times 0.55 = 2.0$$

pounds per square inch, which is pretty close to the limit for good running. If the power be reduced to quarter load, for which a water rate of 18.5 pounds per kilowatt might be expected, then the steam through the nozzles would be reduced to

$$.25 \times 18.5 \div 14.5 = 0.32$$

of that for full load; the fourth stage pressure would in that case be reduced to

$$3.67 \times 0.32 = 1.17$$

of a pound, which is clearly beyond the limit. If the designer considers that it is desirable to have at least 1.5 pounds pressure in the fourth stage for quarter load, he will assign

$$1.5 \div 0.32 = 4.7$$

pounds to that stage for full load. But instead of computing by this or any other way, the designer may conservatively assign 4 pounds for the designed pressure of the fourth stage.

Having decided to assign a penultimate-stage pressure of about 4 pounds, the designer will increase the heat portion for the last stage enough to bring about that condition. For the turbine under discussion he may proceed as follows:

Entropy.	Pressure.	Temperature.	Heat contents.	Difference.
1.63	195.5	380	1271.4	...
1.63	4.0	153	984.6	286.8
1.63	1.0	102	910.4	74.2

$$286.8 \div 4 = 71.7.$$

The entropy table is entered in the usual way with the initial pressure and superheat (or moisture), and in the appropriate

entropy column the heat properties are taken as set down in the table; the first difference is distributed into four stages giving 71.7 B.T.U. per stage; the last difference 74.2 is assigned to the last stage. The work now is that of design with unequal stages which has already been illustrated. It is evident that the velocity of the steam from the last set of nozzles will be greater than from the preceding sets, which will have a slight effect on the efficiency for the last stage.

**Overloads.**—Steam-turbines for electrical work are frequently assigned an overload of twenty per cent of the rated load, and the design is worked out for this maximum normal load. Under the normal working of the turbine up to this maximum load, all the steam is admitted through the first stage nozzles or guides. A turbine designed for such conditions is likely to have the best efficiency at the maximum normal load; but there is little, if any, loss of efficiency at the rated load or at somewhat smaller loads. An electric station has a number of turbines which can be brought into service according to demand so that underloading can be avoided.

To provide for abnormal overloading, it is customary to admit steam directly to some of the lower stages. This is equivalent to admitting steam through a by-pass valve to the low-pressure cylinder of a compound engine; it gives more power but with poor efficiency.

Impulse turbines have overload nozzles supplied to some of the lower stages. They take steam directly from the main steam pipe, and are designed to expand nearly to the normal pressure in the stage to which they are applied. The steam from such overload nozzles has a relatively high velocity, and consequently a reduced efficiency. Overload nozzles will effect efficiency least when applied to an early stage, but are more effective when applied to a later stage. A turbine with few pressure stages may have overload nozzles added to the second set; the Rateau type turbine, described on page 115, has overload nozzles in the third set.

Reaction turbines have steam for overloads admitted through

a by-pass valve to the space between the first and second cylinders.

The control of steam for overload is either by the main governor or by a special governor; impulse turbines are usually of the first type and reaction turbines of the second. The Rice governor, as applied to the Curtis turbine, opens all the normal load nozzles in succession as the load increases, and if there is further demand, it opens the overload nozzles. The Parsons turbine has a special governor which runs idle for normal loads, but opens the overload valve if the speed falls below a determined limit; this occurs when the gusts from the main governor valve follow in such a manner as to give practically a constant blast of full-pressure steam.

Since the use of overload steam is abnormal it need not be considered in the design of the turbine. The economic conditions during overload can best be determined by tests after the turbine is built, and may be inferred by some simple method of comparison from turbines already built, even when conditions of operation are different.

If overload nozzles are added to the second set of the five-stage turbine, discussed on page 171, the heat assignment will be about twice that of normal nozzles and the velocity will be

$$\sqrt{2} = 1.4$$

times the normal velocity. Its loss of efficiency from excess of velocity will probably be not more than two per cent, and this will apply to the steam through overload nozzles only. There will be further loss of efficiency due to derangement of pressures, which will increase in all the stages. The abnormal weight of steam will be passed through the lower sets of nozzles by the combined influence of higher velocity and lower specific volume.

The overload pressures in the various stages may be estimated by the method of page 246, using Rankine's equation. The calculation is the same as that applied to estimating underload

pressures, except that in this case the pressures increase in the stages. It will seldom be worth while to make the step-by-step check calculation. If undertaken the overall heat factor may be reduced a little at discretion, and the steam from the overload nozzles counted with the main supply steam without going into secondary refinements.

**Adding and Withdrawing Steam.** — There are a number of combinations in power development or industrial operations, which result in the normal addition of steam to the turbine at a lower pressure, or the withdrawal of steam from some of the lower stages. When the amount of steam added or withdrawn is a considerable fraction of the entire amount, the turbine must be designed to meet the conditions.

Two distinct problems arise: (1) how shall the design be modified, and (2) what are the economic conditions of the arrangement? The methods of answering these questions will be illustrated by one example for each condition.

**Steam for Heating.** — A condition likely to arise in practice is the withdrawal of steam from a convenient stage of a turbine for heating buildings, the condensed steam being returned to the boilers at a relatively high temperature.

As an illustration, let it be assumed that one-fifth of the normal steam be withdrawn from the eleventh stage of the Rateau turbine, to which the table on page 90 belongs. For sake of simplicity this problem will be computed directly, without considering compromises that may be accepted in practice, because steam for heating is taken only for a part of the year, or any other complications.

In this case the modification of the design may be very simple, especially as the ninth and all following stages have full admission, and the blade lengths are sufficient; namely, we may diminish the radial dimensions of the nozzles and the blade lengths by one-fifth; this will apply to the twelfth and all succeeding stages; for example, the blade lengths for the twelfth stage will be

Entrance 0.96 of an inch, Exit 1.02 inches.

The direct effect of withdrawing one-fifth of the steam from the eleventh stage is that the power of the turbine is reduced in a like proportion. The last thirteen stages lose

$$\frac{1}{5} \times \frac{3}{4} \times 8500 = 921 \text{ shaft horse-power.}$$

If the turbine should be required to deliver the same power under the new conditions, it would be sufficient to increase all the areas of nozzles and blade lengths in the ratio

$$8500 \div (8500 - 921) = 1.12,$$

before making the adjustment just stated.

Under the new condition all the steam still passes through the turbine, and, in a sense, the steam per shaft horse-power per hour is increased in the ratio of the original to the reduced horse-power that is in the ratio

$$8500 \div (8500 - 921) = 1.12.$$

But it would be unfair to charge to the steam heating such a portion of the coal as might be derived from such a ratio, because the condensation from the heating system is returned to the boiler at a relatively high temperature.

The fact that the table on page 88 gives the quality of the steam allows us to state our computation in the form customary for work involving heating; if the heat contents were given the computation would be at least as easy.

We may consider that of each pound of steam supplied to the turbine 0.8 goes directly through; the other 0.2 goes through eleven stages and is condensed in the heating system. We can make computations for these two portions separately, and add the results together, in dealing with the development of power.

The initial temperature is 366° F. and the final temperature of steam exhausted from the turbine is 102°, consequently the heat required in the boiler to heat one pound of water from 102° and to vaporize it at 366° is

$$x_1 r_1 + q_1 - q_2 = 0.999 \times 856.8 + 337.8 - 70 = 1123.8 \text{ B.T.U.}$$

We may assume that the steam which is withdrawn at 21.2 pounds absolute or 6.5 pounds by the gauge is collected at atmospheric pressure and returned to the boiler at 212° F. The heat required per pound in the boiler to revaporize it will be

$$0.999 \times 856.8 + 337.8 - 180.3 = 1013.5 \text{ B.T.U.}$$

The mean heat per pound is, therefore,

$$0.8 \times 1123.8 + 0.2 \times 1013.5 = 1101.7 \text{ B.T.U.}$$

The normal steam consumption per shaft horse-power per hour is given on page 82 as 11.85 pounds; consequently the normal thermal units per shaft horse-power per hour is

$$(x_1 r_1 + q_1 - q_2) 11.85 \div 60 = 1123.8 \times 11.85 \div 60 = 222.$$

The steam consumption per shaft horse-power per hour with steam withdrawn is

$$11.85 \times 8500 \div (8500 - 921) = 13.28 \text{ lbs.},$$

and the corresponding thermal units per minute are

$$1101.7 \times 13.28 \div 60 = 244.$$

The increased cost of power in heat units is

$$244 \div 222 = 1.099.$$

Consequently the cost of power is ten per cent greater, and a corresponding portion of the coal should be charged to heating. If the turbine is increased in size to meet the new duty, then a proportional part of cost, interest, maintenance, and depreciation must be charged in addition; but the bookkeeping at this point is likely to be uncertain.

**Auxiliary Steam to Turbine.** — A considerable portion of all the steam from the boilers of a steamship is used by the auxiliary machinery, such as air pumps, circulating pumps, and feed pumps. Steam for other purposes, and especially for heating, must also be provided. The greater part of the auxiliary machinery does or can work without a vacuum, and there is advantage in returning the steam to a lower stage of a steam-turbine instead

of throwing it into a condenser. There are, however, various ways of utilizing the heat in the steam from auxiliaries, especially in feed-water heaters, so that the study of the proper arrangements for a steamer is one for an experienced marine engine designer.

As an illustration, let it be assumed that all the auxiliary steam is returned to the fifth stage of the marine turbine to which the table on page 195 belongs; the pressure in that stage is 16.5 pounds absolute or 2 pounds above the atmosphere.

It is not easy, even when feed-water measurements are made on a steamer, to separate the steam used by auxiliaries; nor is it easy to determine the power of the auxiliaries or estimate their individual steam consumptions. We may, for a problem, assume that the auxiliaries use fifteen per cent of all the steam from the boilers.

It is convenient to use an overall heat factor for all the auxiliaries. A proper determination would involve the determination or estimate of the power and steam consumption of the individual machines, and a comparison with the steam consumption of Rankine's cycle for each. Let us assume at random that the factor is 0.4.

Since the return of the feed water to the turbine is a normal arrangement it should enter into the design of the turbine; but the design may well be made separately, and then modified.

A question of considerable difficulty is the estimation of the quality of the auxiliary steam as it enters the turbine. For stationary plants it is desirable to pass all such steam through a separator and return the water to the boilers directly; and then the steam from the separator may be taken as dry, the moisture being two per cent or less. The space and complication of a separator and its drain will often exclude it from a marine engine room. A direct estimate may be made by aid of the overall auxiliary heat factor; it will be given for what it is worth, even though the factor was taken at random, and even though there is likely to be loss of heat on the way to the condenser.

The conditions of the auxiliary steam and the computation for the quality are as follows:

	Pressure.	Temperature.	Heat contents, entropy 1.52.	Heat contents, actual	Dryness, factor.
Initial . . . . .	244.1	399	1192.4	1192.4	0.96
Final. . . . .	16.5	218	998.3	77.6 1114.8	
			194.1		

Here the apparent available adiabatic heat at entropy 1.52 from  $399^{\circ}$  to  $218^{\circ}$  is 194.1 B.T.U., and with a heat factor 0.4 the heat actually changed into work in the auxiliaries is 77.6 B.T.U. per pound of steam. This heat subtracted from the initial heat contents gives the quantity 1114.8 as the heat in each pound of steam exhausted from the auxiliaries. Searching the entropy table at  $218^{\circ}$  it appears that this heat contents corresponds with a value of the dryness factor 0.96, so that there appears to be four per cent priming in the auxiliary exhaust.

In comparison the priming in the fifth stage of the turbine is found to be ten per cent, as follows: The heat contents of the steam at  $218^{\circ}$  in column 7 of the table on page 195 is 1048.8 B.T.U., and looking along the line of the entropy table at this temperature the corresponding value of the quality is 0.90, so that there is ten per cent priming.

The resultant quality of the steam after the auxiliary steam has entered the turbine will be

$$0.85 \times 0.9 + 0.15 \times 0.96 = 0.91.$$

To justify this calculation we may add the products of the heat contents, as follows:

$$0.85(x_r + q) + 0.15(x_a + q) = (0.85x_r + 0.15x_a)r + q,$$

in which  $x_r$  is the quality in the turbine before the auxiliary steam enters and  $x_a$  is the quality of that steam.

The conclusion that the admission of the auxiliary steam does not much affect the quality of the steam in the turbine leads to a simple modification of the original design without the auxiliary steam. If we ignore the increased dryness, we may simply increase the area of the nozzles and the blade lengths by dividing them by the factor 0.85. If we desire to allow for the drying of the steam we may further increase those quantities by the ratio of the turbine dryness factor to the resultant factor. In the case in hand we should add about one per cent.

In order to find the influence on economy as compared with a turbine without the arrangement for the auxiliary steam, we may consider that the first five stages have the heat portions (column 2, table on page 195)

$$66.33 + 4 \times 30.82 = 189.6 \text{ B.T.U.}$$

and the remainder have the heat portions

$$30.82 + 8 \times 11.32 = 121.4 \text{ B.T.U.}$$

The auxiliary steam takes the first part of this heat distribution with the factor 0.4, and the second part with the factor for the turbine 0.68; as a matter of fact the lower stages have somewhat better efficiency, but we will not trouble with that detail. The mean heat factor for the auxiliary steam is

$$\frac{0.4 \times 189.6 + 0.68 \times 121.4}{189.6 + 121.4} = 0.51.$$

The mean heat factor for the turbine is, therefore,

$$0.85 \times 0.68 + 0.15 \times 0.51 = 0.65.$$

The steam consumption for the engine and auxiliaries is, therefore (see page 192),

$$12.6 \times 0.68 \div 0.65 = 13.2 \text{ pounds.}$$

Another view of the matter, and perhaps the most useful one, is the gross steam per shaft horse-power per hour; that is to say, the result that would be found if the total steam per hour were divided by the shaft horse-power of the turbine. This may be found by dividing the quantity just computed by the fraction

of the gross power which is developed by the turbine as follows:

$$13.2 \div 0.85 = 15.5 \text{ pounds.}$$

This quantity gives a direct basis for the determination of the boiler power of the ship. Since there was 6000 shaft horse-power assigned to each turbine, the total steam which the boilers must furnish per hour will be

$$12,000 \times 15.5 = 186,000 \text{ pounds.}$$

**Low-pressure Turbines.** — Conditions frequently arise under which steam at or near the atmospheric pressure is exhausted from engines or discharged after certain industrial operations, Less frequently the pressure is higher; as much as fifty pounds by the gauge in the manufacture of rubber.

Steam-turbines can be advantageously designed to use such steam, the large specific volumes being conducive to efficiency and convenience of construction. If the steam has previously been wasted the gain may be very notable.

There is a passing condition of combination of reciprocating engines and low-pressure turbines for power stations, especially when the demand for power has outgrown the capacity. In certain cases the low-pressure cylinder of a triple-expansion engine has been changed for a smaller one, making a three-cylinder compound engine exhausting at about atmospheric pressure. Space has been found for a low-pressure steam-turbine, and the combination has given both more power and better economy. But when new stations are built turbines exclusively are likely to be installed.

As will be explained in the next chapter, there is difficulty in adapting steam-turbines to steamships of moderate power and speed. One method of meeting this condition has been to combine two high-pressure reciprocating engines on two wing shafts, with a single low-pressure turbine on a central shaft. The distribution of power is in such case optional, but is likely to be nearly in three equal parts; if anything, the turbine is given somewhat more than one-third.

**Heat Accumulator.** — One of the first applications of the low-pressure turbines was by Rateau to use the steam discharged at iron and steel works from steam hammers, rolling-mill engines, etc., at atmospheric pressure. These machines are likely to run interruptedly, a condition which was met by Rateau's invention of a heat accumulator.

The simplest form of his accumulator consisted of old rails piled in an old boiler shell, the steam condensing on the iron and reëvaporating as the supply and the pressure varied. But his first design consisted of a vertical plate-iron shell in which was set a stack of shallow cast-iron cups filled with water; the iron and water formed the heat accumulator on which the entering steam condensed and from which steam evaporated.

The best form of the Rateau accumulator appears to be that represented by Fig. 99 in which *CC* are conduits receiving the steam from the high-pressure exhaust. The steam flows out through the perforations in the lower vertical walls of the conduit, and the baffle-plates *PP* induce a rapid circulation of the water. In the intervals when no steam enters the conduits, or when the supply is insufficient, the water boils at falling pressure.

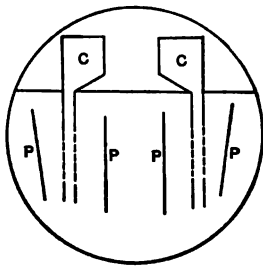


FIG. 99.

An accumulator invented by Schwartz has a vertical shell with a half dozen diaphragms pierced by numerous tubes that project a short distance above the diaphragms. Steam enters at the top and flows down through the tubes and out near the bottom on the way to the turbine. The water that forms the accumulator lies on the diaphragms and at the bottom of the shell; a centrifugal pump draws the water from the bottom and delivers it at the top of the shell from whence it floods the diaphragms.

**Computation for Heat Accumulator.** — The thermodynamic problem for the heat accumulator is the computation of the weight and volume of water required; there are four elements

that enter into this calculation: (1) the weight of steam to be dealt with; (2) the variation of the supply; (3) the allowable fluctuation of pressure, and (4) the weight of water required; the fourth item is usually taken as the quantity sought, though of course any one may be so treated.

*For example*, let us compute the water for a heat accumulator to deal with 30,000 pounds of steam per hour, of which 5000 pounds are delivered regularly, and 25,000 pounds interruptedly. Assume that the latter is delivered regularly for three minutes, and is interrupted for one minute, that is, for one-fourth of the time. Assume further that the pressure may vary from 0.7 of a pound below the atmosphere to 2.3 pounds above; that is, from 14 to 17 pounds absolute.

Now the turbine will use 500 pounds of steam in one minute, and of this one-sixth comes from the regular delivery, leaving

$$\frac{5}{6} \times 500 = 417 \text{ pounds per minute}$$

to be supplied by the water of the accumulator.

The heat of vaporization at 17 pounds is 965, and at 14 pounds is 971.2; the mean value is about 968. Consequently the heat that must be supplied by the heat accumulator during the one minute of interruption is

$$417 \times 968 = 404,000 \text{ B.T.U.}$$

The heat of the liquid at 17 pounds is 187.8, and at 14 pounds is 177.8; so that one pound of water will yield ten thermal units when the pressure is reduced from 17 to 14 pounds absolute. Therefore the weight of water required will be

$$404,000 \div 10 = 40,400 \text{ pounds.}$$

This is about 18 tons of 2240 pounds each.

Taking the weight of the water at 60 pounds per cubic foot, the volume occupied will be

$$40,400 \div 60 = 673 \text{ cubic feet.}$$

To provide for contingencies a safety valve may be provided to discharge into the atmosphere at a pressure of 2.3 pounds by the gauge, or at such higher pressure as the designer may

choose. On the other hand, it is customary to provide for delivering steam from some independent source, either to the accumulator or directly to the turbine, if the exhaust supply is deficient or unduly interrupted.

**Mixed-Pressure Turbines.** — When the steam supply to a low-pressure turbine, from high-pressure engines or other sources, is liable to be irregular or insufficient, the most ready way of supplying the deficiency to impulse turbines is through high-pressure nozzles directly from a steam main. Such turbines are called mixed-pressure turbines. The velocity of the steam from the high-pressure nozzles is very high and the blades receiving it must be strong and well secured; they may be made of steel or high-grade bronze.

To meet a similar condition, a Parsons turbine at a steel works was designed to run either with high-pressure steam from boilers, or low-pressure steam exhausted from high-pressure machinery. The design, in general appearance, differs from the usual form, only in the provision for a free supply of the exhaust steam to the space leading to the third or large cylinder. When this steam at atmospheric pressure is sufficient, the high-pressure steam is shut off and the high-pressure end of the rotor runs in steam at atmospheric pressure; when all the steam comes from the boiler, the pressure at entrance to the low-pressure cylinder is eight pounds absolute, and the turbine generates about the same amount of power. The turbine can run with steam from both sources.

**Compound Turbines.** — In a certain sense all turbines except simple impulse turbines, like the de Laval, are compounded; but for certain purposes, especially for marine propulsion, it has been found convenient to pass steam in succession through separate turbines. The Parsons turbine, which was the pioneer for this purpose, is always so treated. For powers not exceeding 10,000 horse-power, the Parsons turbines are set on three shafts; for very large powers on four. In the first case there is one high-pressure turbine and two low-pressure turbines; the high-pressure turbine develops about one-third the power and the two low-

pressure turbines the other two-thirds. When four turbines are used there are two distinct trains; that is, there is one starboard high-pressure turbine exhausting into a starboard low-pressure turbine, and a similar arrangement on the port side.

The turbines are usually intended to run at the same number of revolutions, so the design of a train of turbines does not differ essentially from that of a turbine with two or more cylinders on a single rotor. There is an appreciable loss of pressure between the turbines of a train for which allowance should be made; if this is done then the design should be for each turbine separately, the low-pressure turbine (or turbines) taking steam at the estimated pressure and quality from the high-pressure turbine.

To illustrate this phase, suppose that there is a loss of four pounds pressure between the central high-pressure turbine and the two wing turbines, when the steamer has three shafts. Take an even distribution of power, with the initial pressure of 150 pounds gauge, and a vacuum of 28 inches of mercury. Take an overall heat factor of 0.6 and treat the turbines as two stages, one having one-third of the power and the other two-thirds.

The initial conditions lead to the entropy 1.56 for saturated steam, at which the calculation can be made as follows:

Pressure.	Temperature.	Heat contents.
164.8 1.0	366 102	1193.3 871.1
		322.2

The ratio for temperature distribution is 1.08, and the factors may be taken from Fig. 100 as set down in column 2. The calculation may be made as follows:

Heat portions.	Factor.	Heat assignments.	Heat contents.	Pressure.	Temperature.
...	...	...	1193.3	164.8	366
107.4	1.053	113.2	1080.1	37.6	263.5
214.8	0.973	209.0	871.1	1.0	102

If the final temperature be taken as  $266^{\circ}$  for the high-pressure turbine, and  $259^{\circ}$  be taken for the initial temperature of the low-pressure, then the drop of pressure will be

Temperature.	Pressure.	Difference.
266	39.2	...
263.5	37.6	1.6
259	34.8	2.8

which is about right, as there are two low-pressure turbines.

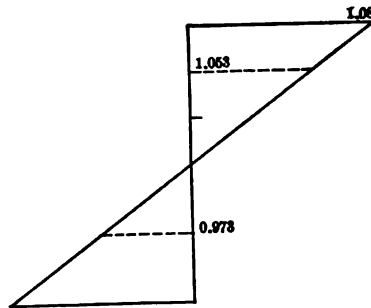


FIG. 100A.

**Combination Turbines.** — Propositions have been made from time to time, and in a few instances have been put into effect, to combine the impulse turbine with the reaction turbine. For example, the Westinghouse Company offers for marine work, a turbine which has a single stage repeated-flow turbine, followed by nine barrels of the Parsons type having two rows of blades for each barrel; this is in combination with a proposal to use gears for reducing revolutions, but that is not a necessary feature of the arrangement of the turbine.

The Curtis marine type as illustrated by Fig. 191, page 192, has a varying number of rows of blades per stage; it has been proposed to use only one row of blades in the drum stages. Clearly the transition to the reaction type for the drum would be facile if desired.

## CHAPTER XI

### MARINE STEAM-TURBINES

THE customary and desirable arrangement of steam turbines for marine propulsion places the turbine on the same shaft as the propeller. It is customary to use two, three, or four propellers; but even so, desirable conditions would lead to the use of a comparatively large propeller having slow rotation, while the steam-turbine should have a high peripheral speed and a small diameter. A compromise is made, taking as small a propeller as possible running at a high speed, and the turbine must then be given such dimensions as will meet the conditions imposed. Though the Parsons turbine has been most extensively used, and has afforded the most experience, its conditions emphasize the difficulty of a satisfactory design, and have probably tended to the use of small and inefficient propellers. ✓

The selection of the proper propellers for a ship, and especially for a turbine steamer, where efficiency must be sacrificed to some extent, requires a thorough knowledge of the characteristics of propellers and the problems of marine propulsion. A comprehensive treatment of these matters will be found in a textbook\* by the writer, or in the works of Naval Constructor D. W. Taylor, † U. S. N. Those particular features which affect the design of propellers for turbine steamers will be stated briefly here. The article by Mr. Speakman, already used extensively in Chapter VIII, will be again quoted freely.

**Cavitation.** — The feature which fixes a fairly definite limit to the reduction of size and increase of speed of propellers is known as cavitation, a term introduced by Mr. S. W. Barnaby, ‡

\* Naval Architecture, John Wiley & Sons.

† Speed and Power of Ships, John Wiley & Sons.

‡ Marine Propellers.

who first gave a clear statement of the phenomenon, and proposed a method of avoiding it.

Without going into details, it may be said that the screw propeller can exert a thrust to drive the ship, because it imparts acceleration to a stream of water. When an attempt is made to exert too much thrust by a propeller, the stream of water becomes broken into eddies, and the propeller can neither absorb the power nor exert the thrust that might be expected, from an observation of its performance at different revolutions, increasing up toward the place at which the failure occurs.

Our systematic information concerning the characteristics of propellers comes from experiments on model propellers, none of which is more than two feet in diameter; more commonly the diameter of a model propeller has been 12 or 16 inches. From the results of such experiments the performance of full-sized propellers may be inferred with a fair degree of satisfaction, except in case of the phenomenon called cavitation. Normal forms of model propellers cannot be made to cavitate, and while valuable information can be deduced from the cavitation of models having abnormal forms that can produce cavitation, that information cannot be used directly to predict conditions producing cavitation. As a consequence, our knowledge of cavitation is drawn from observation of conditions where cavitation has inadvertently been experienced. Since the advent of very fast craft, like torpedo boats, and especially since the introduction of steam-turbines, builders have had much trouble from cavitation, but most commonly the appearance of cavitation with a certain propeller has led to its immediate rejection, and if builders have acquired further information but little has been published.

Practically we have a theory and method proposed by Barnaby and the report by Speakman from a number of turbine steamers, together with a suggestion by the latter concerning the peripheral speed of propeller blades. Whether or not the conceptions proposed are accepted as a complete explanation of cavitation, the

relation of those conceptions to the performance of turbine steamers is our best guide concerning cavitation.

**Propeller Thrust.** — The shaft horse-power applied by turbines to propulsion of a ship is determined by aid of torsionmeters, and is fairly well known, as is also the speed of the ship. From them an estimate may be made of the thrust applied by a turbine to driving the ship.

For this purpose the shaft horse-power is first to be multiplied by the efficiency of the propeller, to determine the power actually applied to propulsion. The propeller efficiency is taken from experiments on model propellers and for them is well known. The actual efficiency of the full-sized propeller behind the ship is, however, not well known. Two elements affect it: (1) the efficiency of a large propeller is probably a little less than the model propeller; (2) the propeller gains an advantage from working in the wake behind the ship and can abstract some energy from it.

Now the efficiency of a quick-running propeller of the type used for turbine steamers is about 55 per cent, as determined from models. The combined influence of the two elements mentioned is to increase the efficiency of the propeller in place, in some cases by as much as 5 per cent, additively. But for turbine steamers the increase is uncertain and likely to be small. For our present purpose we may take 55 per cent for the efficiency in all cases.

The speed of a ship is usually given in knots (6080 feet) per hour; for inland navigation the statute mile (5280 feet) is used.

To compute the thrust we will first reduce the horse-power applied to propulsion to work in foot-pounds per minute, and then divide by the speed of the ship in feet per minute. This may be expressed by the following equation when the speed is in knots.

$$\text{Thrust} = \frac{0.55 \text{ S.H.P.} \times 33,000}{\frac{6080}{60} V} \quad . \quad . \quad . \quad (1)$$

in which S.H.P. stands for the shaft horse-power for one turbine, and  $V$  is the speed in knots per hour.

*For example.* — The torpedo-boat destroyer *Terry*, for which data are given on page 275, had a speed of 30 knots an hour, and developed 13,350 shaft horse-power with three turbines. The thrust for one turbine may be taken as

$$\text{Thrust} = \frac{0.55 \times \frac{1}{3} \times 13,350 \times 33,000}{\frac{8.080}{8.0} \times 30} = 26,600 \text{ pounds.}$$

**Projected Area.** — The projected area of a propeller is the area of the projection of all the blades (exclusive of the hub) on a plane perpendicular to the shaft.

The projected area is commonly given by the draughtsman. When the so-called developed area, or the helicoidal area, is given the projected area may be computed by the equation

$$\text{Projected area} = \frac{\text{developed area}}{\sqrt{1 + 0.425 (\text{pitch-ratio})^2}}, \quad \checkmark$$

in which the pitch-ratio is the ratio of the pitch of the helical surface (or screw) to the external diameter. It may be said that the helicoidal area is the true area of the helical surface of the blades and the developed area is an approximation thereto. This equation can be used for blades which have an elliptical contour, which form is habitual for turbine propellers.

**Barnaby's Method.** — To apply Barnaby's method divide the thrust calculated by equation (1), by the projected blade area in square inches; should the result be equal to or greater than 11.25 pounds per square inch, it is considered that cavitation is liable, the immersion of the propeller being one foot. If the immersion is greater than one foot the figure 11.25 may be increased by three-eighths of a pound for each additional foot.

Mr. Barnaby bases this method on a theory that cavitation is produced by too great a reduction of pressure in the water approaching the propeller. It is not certain that this is the principal reason for cavitation; but his method provides that blades shall be wide and sharp at the edge as usually made, and those conditions are favorable.

*Example.* — The propellers of the *Terry* had each 12.98 square feet or 1870 square inches projected area, consequently with 26,600 pounds thrust, already computed, the thrust per square inch may be taken as

$$26,600 \div 1870 = 14.2 \text{ pounds.}$$

**Propeller Peripheral Speed.** — In the paper already quoted, Speakman advises that the peripheral speed of the tips of the blades of a propeller shall not exceed 12,000 feet per minute, if cavitation is to be avoided, and further recommends that the speed be reduced to 9000 feet per minute when practicable.

He further says that the thrust per square inch by Barnaby's method is likely to be a pound per square inch for each 1000 feet peripheral speed; but this does not appear to be more than a general indication.

*Example.* — The propellers of the *Terry* were 5.25 feet in diameter, and made 834 revolutions per minute; the peripheral speed was therefore about

$$\pi 5.25 \times 834 = 14,000 \text{ ft. per minute.}$$

Though both computations of thrust and peripheral speed for the *Terry* indicate that cavitation is to be expected, there is no evidence from the speed and power trials to show that it occurred. The immersion of the tips of the propellers was more than a foot, but the proper figure cannot be given.

**Balancing Propeller Thrust.** — It is customary to balance the thrust of the propeller by steam pressure on the turbine. The method can be illustrated by a computation for the Curtis marine type to which the table on page 195 belongs.

The shaft horse-power for one turbine was assumed to be 6000, and we may now assign 33 knots per hour for the speed of the ship. With these conditions the thrust will be

$$\frac{0.55 \times 6000 \times 33,000}{\frac{2.080}{80} \times 33} = 32,500 \text{ pounds.}$$

If, further, we assign 72 inches for the pitch diameter of the turbine and 20 inches for the effective diameter of the shaft

at the drum, we may proceed to compute the desired pressure at the front end of the drum as follows: The diameter of the drum at the seventh set of nozzles may be taken as 70 inches, and at the thirteenth set of nozzles 60 inches; the effective areas at the front end will be

$$\frac{\pi 70^2}{4} - \frac{\pi 20^2}{4} = 3848 = 314 - 3534 \text{ square inches,}$$

and at the after end

$$\frac{\pi 60^2}{4} - \frac{\pi 20^2}{4} = 2827 - 314 = 2513 \text{ square inches.}$$

The effective after area is exposed to the pressure in the last stage, which is one pound absolute. The resultant pressure of 2500 pounds will tend to force the rotor forward, and must be added to the propeller thrust, making the sum

$$32,500 + 2500 = 35,000 \text{ pounds.}$$

This must be balanced by the pressure in the sixth stage, and the absolute pressure required will be

$$35,000 \div 3500 = 10.$$

which chances to be the pressure computed for that stage.

Such an absolute compensation cannot be expected, nor is it desirable; there is reason for designing for a considerably higher pressure at the drum so that the thrust will be overbalanced, and there will be a pull on the thrust block at all times. Some designers overbalance to the extent of one-third. In the first place the computation of the thrust is uncertain, and so also is the computation of pressure though to a less extent, and a margin can be added by the design to cover the uncertainty; and secondly, the pressures in the stages decrease more rapidly for reduced powers than does the thrust, and to be sure that there will always be a pull on the thrust block, the full power thrust must be overbalanced. An experienced designer, by study of results from tests on turbines already built, may be able to determine the resultant effect of these influences. The reason for over-balance-

ing, and thus working with a pull on the thrust block, is that the backing effect of the propeller cannot be compensated in the type of turbine under consideration, and as it is a pull, the ensuring of a pull at all times, both going forward and backing, will avoid shifting the thrust-block pressure from one side to the other of the bearing rings. In particular, it is very annoying to have the pressure on the thrust block shift from a pull to a thrust, while the turbine is changing power.

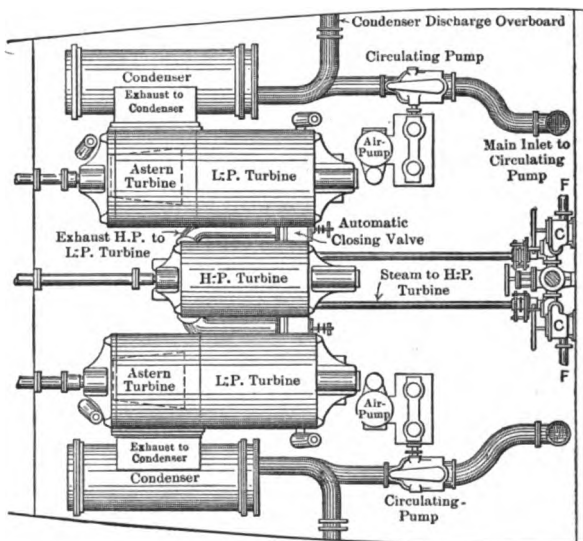
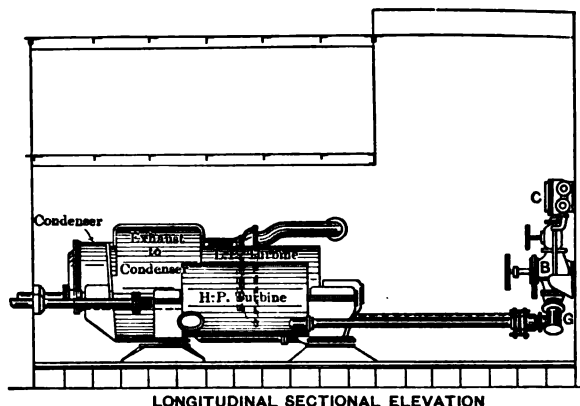
**Turbine Peripheral Speed.** — A propeller shaft should be horizontal, since an inclination downwards towards the propeller tends to produce vibration from the inclined action of the propeller blades on the water; this is especially important for turbine ships on account of the rapid rotation. Even though the turbine propellers are relatively small, it will be found that after they have been given sufficient immersion there will often be scant spaces for the turbine between the shaft and the floors of the ship. The arrangement must be worked out for each case individually, but in a general way it may be said that the pitch diameter of a steam turbine will not be greater than the diameter of the propeller, and frequently is less. In the case of the *Terry*, the greatest pitch diameter of the low-pressure turbine (Parsons type) is  $46\frac{1}{2}$  inches; the diameter of the propeller is 5 feet or 60 inches, and consequently the peripheral speed of the turbine is

$$13,000 \times 46.5 \div 60 = 10,000$$

feet per minute or 167 feet per second. This is the greatest speed, and the other stages will have less. A table of peripheral speeds has been given on page 204.

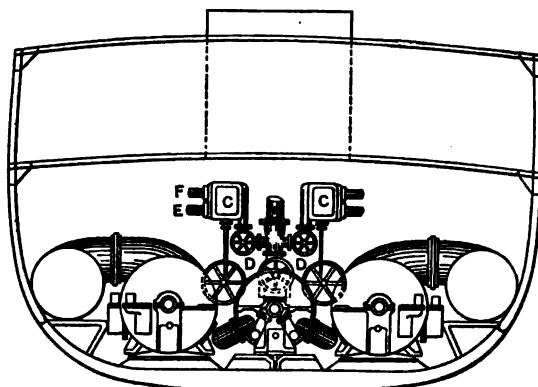
**Backing Turbines.** — Special turbines must be provided for backing; they are usually designed to give 0.6 of the full power ahead, but as turbine ships are high-powered this is sufficient. By some sacrifice of efficiency they may be made small and compact; they are placed at the exhaust end of the low-pressure turbine and run in a vacuum when idle, so that there is little resistance.

**Engine Room.** — The arrangement of the engine room with three Parsons turbines is shown by Figs. 100 and 101; this



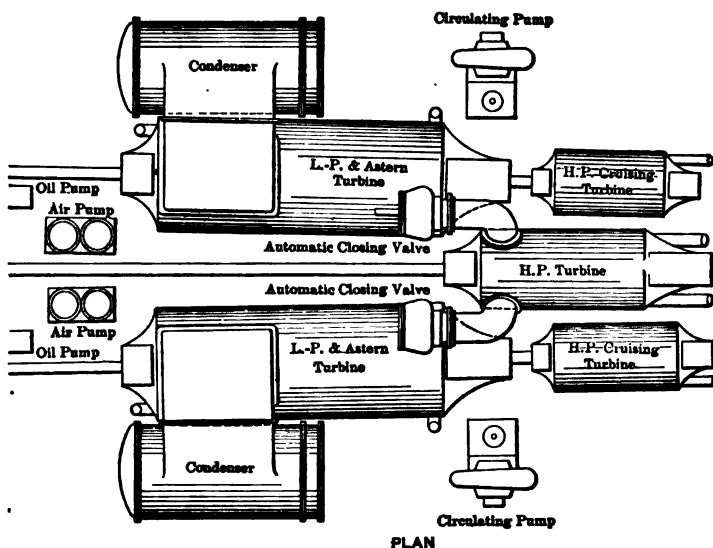
**FIG. 100.**

is typical for ships of moderate size; for larger ships there may be four turbines, the high-pressure turbines being in the middle.



CROSS-SECTIONAL ELEVATION

FIG. 101.



PLAN

ARRANGEMENT OF TURBINE MACHINERY FOR A CRUISER

FIG. 102.

An arrangement for a cruiser is shown by Fig. 102, which differs mainly in the addition of two turbines for cruising, when power and speed are much reduced. Steam is admitted first to one auxiliary high-pressure cruising turbine on one wing shaft, and from thence goes to a second or low-pressure cruising turbine on the other wing shaft; after passing these turbines the steam passes to the full power turbines.

When Curtis turbines are used there are commonly only two, though recent designs by the Fore River Company provide for three turbines. In any case each turbine is complete in itself as shown by Fig. 72, facing page 190.

**Destroyer Turbines.** — A recent publication gives exceptionally full particulars of the turbines and performances of four U. S. torpedo-boat destroyers, *Sterett*, *Perkins*, *Roe*, and *Terry*; the first two have Curtis turbines, and the other two, Parsons turbines; descriptions are given by Captain W. N. Little,\* U. S. N., and Mr. C. F. Bailey.\*

## PARTICULARS OF HULL

Length between perpendiculars, feet . . . . .	289
Draught, feet and inches . . . . .	8-4
Displacement, tons . . . . .	742
Coefficient of fineness . . . . .	0.41

Propellers.	Sterett.	Perkins.	Roe and Terry.
Number . . . . .	2	2	3
Diameter, feet and inches . . . . .	6-6½	6-7	5-3
Pitch, feet and inches . . . . .	6-3.6	6-4.3	4-10
Projected area, all blades, square feet . . .	18.2	19.2	12.98

## Immersion, inches:

Center . . . . .	53
Wing . . . . .	36.5

\* Jour. Am. Soc. Nav. Engrs., Feb., 1911.

## TURBINE DATA (STERETT AND PERKINS)

Stages.	Nozzles.			Guides, no. rows.	Blades.		
	Number.	Total area, sq. in.	Arc of nozzles, degrees.		No. rows.	Width, inches.	Length, inches.
1	19	8.89	43	3	4	1 to 0.75	1.33 to 3.00
2	33	15.15	66	2	3	0.75	1.54 to 2.70
3	45	20.66	90	2	3	0.75	1.54 to 2.70
4	65	29.84	130	2	3	0.75	1.54 to 2.70
5	95	43.61	190	2	3	0.75	1.54 to 2.70
6	140	64.27	280	2	3	0.75	1.54 to 2.70
7	174	101.6	348	1	2	0.75	1.85 to 2.38
8	136	126.5	360	1	2	0.75	2.29 to 2.94
9	136	165.3	360	1	2	0.75	2.99 to 3.84
10	136	211.7	360	1	2	0.75	3.82 to 4.92
11	136	276.3	360	1	2	0.75	4.99 to 6.42
12	136	361.3	360	1	2	0.75	6.53 to 8.40
13	136	479.7	360	1	2	1.00	7.93 to 8.92
14	136	619.8	360	1	2	1.00	8.09 to 9.10

The nozzle angles are not given in the report; for the first six stages the angle is probably  $20^{\circ}$ ; for the drum stages the angle is probably larger, perhaps  $25^{\circ}$  to  $30^{\circ}$ .

The blade angles are not given in the report. It is probable that for all the blades and guides, except those of the last two barrels of the low-pressure main turbine, and the last row of the astern turbine, the admission angle at the back of the blade is about  $60^{\circ}$ , and that the exit angles are about  $20^{\circ}$ . The last three barrels of the low-pressure turbine, as shown in the table, have blades of the same length; consequently the provision for expansion must be by opening the angles, which also increases the velocity. To properly take steam from the sixth stage, the entrance angle to the seventh stage might be kept the standard angle ( $60^{\circ}$ ), and the exit angle might be increased to  $30^{\circ}$  or more. The eighth stage probably had both entrance and exit angles increased; the former might be made  $70^{\circ}$  and the latter  $40^{\circ}$  or  $50^{\circ}$ . In somewhat the same way the blade lengths for the third and fourth barrels of the astern turbine are the same in the table; the latter, therefore, must have had larger angles, which might be like those for the seventh stage of the low-pressure turbine.

## TURBINE DATA (ROE AND TERRY)

Barrel.		Ahead.				
		H.P.C.	I.P.C.	M.H.P.	L.P.	Astern.
1	Diameter of cylinder, inches . . . . .	30 $\frac{1}{2}$	30 $\frac{1}{2}$	32 $\frac{1}{2}$	46 $\frac{1}{2}$	31 $\frac{1}{2}$
1	Length of cylinder, this diameter, inches . .	16 $\frac{1}{2}$	16 $\frac{1}{2}$	13 $\frac{1}{2}$	7	9 $\frac{1}{2}$
1	Rows of blading . . . . .	18	16	10	5	8
1	Length of blades, inches . . . . .	0 $\frac{1}{2}$	0 $\frac{1}{2}$	1 $\frac{1}{2}$	1 $\frac{1}{2}$	0 $\frac{1}{2}$
1	Thickness of calking sections, c. . . . .	120	120	130	130	120
2	Diameter of cylinder, inches . . . . .	31	31 $\frac{1}{2}$	34	48	32 $\frac{1}{2}$
2	Length of cylinder, this diameter, inches . .	16 $\frac{1}{2}$	20 $\frac{1}{2}$	13 $\frac{1}{2}$	7 $\frac{1}{2}$	11 $\frac{1}{2}$
2	Rows of blading . . . . .	18	16	10	5	8
2	Length of blades, inches . . . . .	0 $\frac{1}{2}$	1 $\frac{1}{2}$	2 $\frac{1}{2}$	2 $\frac{1}{2}$	1 $\frac{1}{2}$
2	Thickness of calking sections, c. . . . .	120	130	130	130	130
3	Diameter of cylinder, inches . . . . .	31 $\frac{1}{2}$	32	35 $\frac{1}{2}$	49 $\frac{1}{2}$	35
3	Length of cylinder, this diameter, inches . .	17 $\frac{1}{2}$	20 $\frac{1}{2}$	17	7 $\frac{1}{2}$	14 $\frac{1}{2}$
3	Rows of blading . . . . .	18	16	10	5	8
3	Length of blades, inches . . . . .	0 $\frac{1}{2}$	1 $\frac{1}{2}$	3	3	2 $\frac{1}{2}$
3	Thickness of calking sections, c. . . . .	120	130	240	130	240
4	Diameter of cylinder, inches . . . . .	...	...	38	52	35
4	Length of cylinder, this diameter, inches . .	...	...	18 $\frac{1}{2}$	8 $\frac{1}{2}$	15
4	Rows of blading . . . . .	...	...	10	5	8
4	Length of blades, inches . . . . .	...	...	4 $\frac{1}{2}$	4 $\frac{1}{2}$	2 $\frac{1}{2}$
4	Thickness of calking sections, c. . . . .	...	...	240	130	242
5	Diameter of cylinder, inches . . . . .	...	...	...	54	...
5	Length of cylinder, this diameter, inches . .	...	...	...	8	...
5	Rows of blading . . . . .	...	...	...	4	...
5	Length of blades, inches . . . . .	...	...	...	5 $\frac{1}{2}$	...
5	Thickness of calking sections, c. . . . .	...	...	...	240	...
6	Diameter of cylinder, inches . . . . .	...	...	...	59	...
6	Length of cylinder, this diameter, inches . .	...	...	...	8 $\frac{1}{2}$	...
6	Rows of blading . . . . .	...	...	...	4	...
6	Length of blades, inches . . . . .	...	...	...	7 $\frac{1}{2}$	...
6	Thickness of calking sections, c. . . . .	...	...	...	240	...
7	Diameter of cylinder, inches . . . . .	...	...	...	59	...
7	Length of cylinder, this diameter, inches . .	...	...	...	8	...
7	Rows of blading . . . . .	...	...	...	4	...
7	Length of blades, inches . . . . .	...	...	...	7 $\frac{1}{2}$	...
7	Thickness of calking sections, c. . . . .	...	...	...	241	...
8	Diameter of cylinder, inches . . . . .	...	...	...	59	...
8	Length of cylinder, this diameter, inches . .	...	...	...	8	...
8	Rows of blading . . . . .	...	...	...	4	...
8	Length of blades, inches . . . . .	...	...	...	7 $\frac{1}{2}$	...
8	Thickness of calking sections, c. . . . .	...	...	...	242	...

Diameter of rotor drums: H.P.C., 30 inches; I.P.C., 29 inches; M.H.P., 29 $\frac{1}{2}$  inches; L.P., 43 $\frac{1}{2}$  inches.

## PERFORMANCE FOUR-HOUR FULL-SPEED TRIALS

	Sterett.	Perkins.	Roe.	Terry.
Steam pressure, pounds,				
Boiler, gauge . . . . .	263	257	239	250
Main steam pipe, gauge . . . . .	250	238	233	250
Auxiliary exhaust, gauge . . . . .	13.6	6.2	13	14
H.P. turbine, absolute . . . . .	246 St.	246	217	229
L.P. turbine, starboard . . . . .	241 P.	...	47	55
L.P. turbine, port . . . . .	...	...	52	54
Vacuum, inches mercury,				
Starboard condenser . . . . .	26.8	28.0	27.5	27.5
Port condenser . . . . .	27.7	27.6	27.5	27.0
Barometer . . . . .	29.8	30.1	30.0	30.0
Revolutions per minute,				
Starboard . . . . .	645	596	782	839
Center . . . . .	...	...	790	811
Port . . . . .	618	592	812	852
Average . . . . .	631	594	795	834
Speed, knots . . . . .	30.4	29.8	29.6	30.5
Shaft horse-power				
Starboard . . . . .	...	...	3597	4407
Center . . . . .	...	...	4536	4807
Port . . . . .	...	...	3927	4136
Total . . . . .	12,789	11,668	12,061	13,350
Auxiliary indicated horse-power . . . . .	...	...	507	600
Steam per hour, pounds, all machinery . . . . .	190,017	169,083	199,500	206,850

Details of construction are given by Fig. 103 for the *Roe* and *Terry*.

**Engines and Turbine.** — A combination of two reciprocating engines driving wing screws, and a low-pressure turbine on a central shaft, has been adapted by Messrs. Denny of Dumbarton, Scotland, to the *S. S. Otaki* in the New Zealand trade. This adaptation is specially interesting because comparison is made with the performance of a ship on the same line with ordinary reciprocating engines, namely the *Orari*. The ship could be maneuvered both ahead and backing with the engines only, the turbine being thus cut out.

The *Otaki* had a length of 464½ feet and a displacement of 11,716 tons on a draught of 27½ feet; the *Orari* was 4½ feet shorter.

The engine dimensions were as follows:

	Otaki.	Orari.
Diameter, high pressure, inches . . . . .	24 $\frac{1}{2}$	24 $\frac{1}{2}$
Intermediate, inches . . . . .	39	41 $\frac{1}{2}$
Low pressure, inches . . . . .	58	69
Stroke, inches . . . . .	39	48

The diameter of the turbine rotor was 7 $\frac{1}{2}$  feet, the blade lengths varying from 4 $\frac{3}{4}$  to 12 $\frac{1}{8}$  inches. The revolutions at full power were about 224 while the engine ran at 103.

The absolute pressures in the engine cylinders and the turbine are as follows (the former from indicator diagrams and the latter from mercury columns):

Reciprocating engine.			Turbine.		
H.P., initial.	M.P., initial.	L.P., initial.	Final.	Initial.	Final.
196	91.5	36	11.5	9.8	1.0

The comparison of economy of the two types of propulsion is difficult and uncertain, because the shaft horse-power of the reciprocating engines could not be estimated satisfactorily, nor could the turbine power be determined otherwise than as shaft horse-power. But as the builders had experimented with models of the ships and their propellers, and had tested both types for steam consumption, they estimated the advantage of the turbine at full power to be 17 per cent. The full-power speed was 14.6 knots per hour, while the service speed was about 12 knots; at the latter speed the advantage of the turbine in coal consumption on a voyage was from 12 to 15 per cent, which was considered satisfactory, especially as the turbine was expected to be most advantageous at high power.

An item of interest is that the relatively low speed of the turbine allowed of ordinary sight lubrication of the bearings, and the shaft glands were packed with soft packing in two sections, with water between to form a water-seal.

**Geared Turbines.** — In order to adapt turbines to the propulsion of slow ships attempts have been made to reduce the speed by gearing, and so use a small, light turbine and a large, slow propeller. In order to ensure smooth running it has been considered necessary to use double-helical gears similar to those used for the de Laval turbine. The gears have small pitch and must, therefore, be nicely adjusted to have the driving pressure distributed uniformly over the axial width of the gear, which is considerable.

Such helical gears are very efficient when flooded with oil under pressure; the friction is less than two per cent.

**Parsons Geared Turbines.** — In order to test geared turbines for marine work the Turbinia Company purchased the S. S. *Vespasian*, a freighter built in 1887 to the following dimensions: length 275 feet, beam 38 feet 9 inches, draft 19 feet 8 inches, displacement 4350 tons.

The ship when purchased had the usual type of triple-expansion reciprocating engines with cylinders  $22\frac{1}{4}$ , 35, and 59 inches in diameter, and a stroke of 42 inches. The propeller was of cast iron with four blades, and 14 feet in diameter, and 16.35 pitch.

The ship was first put in good condition and tested with the original engines in place; then the engines were removed and turbines were substituted, the boilers and the propeller remaining unchanged; after the change the ship was again tested to give a basis of comparison.

On the propeller shaft a pair of large helical gears were fixed, the diameter being 8 feet  $3\frac{1}{4}$  inches, and the face of each gear was 12 inches; the number of teeth was 398, and the pitch  $0.7854$  of an inch; the helical angle was  $20^\circ$ . The pinions were 5 inches pitch diameter so that the ratio was 19.9.

There were two turbines in service — a high-pressure turbine on the starboard side, and a low-pressure turbine on the port side — both geared to the wheel on the propeller shaft. The reversing turbine was placed in the low-pressure end of the low-pressure turbine.

The feed water during each test was measured in tanks, and from the tests the following table of results is quoted:

	Engine.	Turbine.
Revolutions of propeller . . . . .	70	71.3
Speed, knots . . . . .	10.2	10.5
Power, indicated, or shaft . . . . .	993	980
Boiler pressure, gauge . . . . .	150	140
Vacuum, inches . . . . .	26.5	28.7
Steam per horse-power per hour . . . . .	18	14.8

The ship with the turbines made 10.7 knots at 73.3 revolutions, and used 14.8 pounds of steam per shaft horse-power.

The gears were reported to work with no trouble and little noise. It is to be noted that at the highest speed the gears transmitted 1095 horse-power at a velocity of about 1900 feet per minute at the pitch circle of the gears; as there were two gears, half of this, or 550 horse-power, was transmitted by each pinion. This is about twice what is done by the de Laval turbine.

**Melville and MacAlpine Gear.** — A very large helical gear for marine propulsion, invented by Admiral Melville and Mr. MacAlpine, has been built by the Westinghouse Company, and tested in the shop when transmitting 7000 horse-power through one double-helical pinion. The turbines and gears for a naval collier have been installed to give practical trial of this type.

The peculiarity of this gear is that the pinion is carried by a floating frame which is capable of a slight rocking motion in a vertical plane, but is held up to the proper position for engagement of the teeth. The frame, which is a rigid casting, is supported at the middle by a transverse plate that has some flexibility and acts like a knife edge of a balance, to weight the tooth pressure at the two ends of the pinion; and should one pressure exceed the other the frame shifts till equilibrium is obtained.

**Electric Drives.** — A number of propositions have been made to drive ships electrically, the current being generated in a set of turbines and generators of much the same type as is used for land installations; the current to be used in motors on the pro-

pellor shafts. It is expected that the better steam economy of such an installation will more than offset the electrical losses. And further, it is expected that by combinations a high cruising economy can be secured for warships, the notable defect of the usual turbine installation being the poor economy at low power and speed. In order to attain a good result from any electrical drive it is necessary to use alternating-current machinery and high voltage — conditions that have been entirely mastered on shore, and which appear to present no unsurmountable difficulty at sea.

A system invented by Mr. W. L. R. Emmet is now under installation in a naval collier, in which its adaptability can be tested but not its real merits, that would appear in a warship, which must cruise at low speed and be capable of high speed, and ought to have good economy under both conditions. Station keeping in squadron is attained by controlling the steam supply of the turbines and thus the voltage.

Some boats for special purposes have electric drives, notably certain fire boats which have turbines of ordinary stationary type directly connected to compound centrifugal pumps. The shafts also carry electric generators which drive the propellers through motors; direct current is used much as in street-car service.

DPYK

8. T  
950

100

200

300

400

0°

8°

6°

5°

6

7°



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